NUMERICAL STUDY OF LAMINAR NATURAL CONVECTION IN AN ENCLOSURE WITH CONJUGATE HEAT TRANSFER

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NUMERICAL STUDY OF LAMINAR NATURAL CONVECTION IN AN ENCLOSURE WITH CONJUGATE HEAT TRANSFER

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by

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CERTIFICATE

This is to certify that the thesis entitled **Numerical study of laminar natural convection in an enclosure with conjugate heat transfer**, submitted by **Prasad P. Wadile** to National Institute of Technology, Rourkela, is an authentic record of bona fide research work carried under my supervision and I consider it worthy of consideration for the award of the degree of Master's of Technology of the Institute.

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DECLARATION

I certify that

1. The work contained in the thesis is original and has been done by myself under the general supervision of my supervisor.
2. The work has not been submitted to any other Institute for any degree or diploma.
3. I have followed the guidelines provided by the Institute in writing the thesis.
4. Whenever I have used materials (data, theoretical analysis, and text) from other sources, I have given due credit to them by citing them in the text of the thesis and giving their details in the references.

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This project would not be completed without all the help from my lab-mates. The time we spent together other than research work was truly refreshing. Any small technical talks that we had together were surely helpful for improving this thesis.

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LIST OF SYMBOLS AND ABBREVIATIONS

L	Length of the side of the enclosure (side and top)
t_w	Thickness of the solid wall (side and top)
t_i	Thickness of the insulation
t_c	Thickness of the cavity above insulation
k_a	Thermal conductivity of fluid i.e air
k_i	Thermal conductivity of insulation material, k_a=k_i
k_w	Thermal conductivity of solid wall
K_r	Thermal conductivity ratio of solid wall and fluid,k_w/k_f
p	Pressure
P	Dimensionless pressure, $p/\rho U_0^2$
U_o	Average velocity, $\sqrt{g\beta L(T_H - T_C)}$
T_c	Low temperature, cold
T_H	High temperature, hot
T_a	Atmospheric air temperature
T_w	Temperature on the outer side of the right side wall
c_p	Specific heat of the fluid
g	Acceleration due to gravity
Pr	Prandtl number of fluid,$\frac{\nu}{\alpha}$
Gr	Grashof number,= $g\beta L^3 \triangle T/\nu^2$
Re	Reynolds number,$\rho U_0 L/\mu$
Ri	Richardson number, $\frac{Gr}{Re^2}$
K^*, ρ^*, c_p^*	Dimensionless numbers used for computation.
$\overline{u}, \overline{v}$	Dimensional mean velocities in x, y-directions respectively
U, V	Non-dimensional velocities in X, Y-directions respectively
x, y	Dimensional co-ordinates
X, Y	Non-dimensional co-ordinates

Greek symbols

α	Thermal diffusivity, $^{k}/_{\rho c_p}$
β	Co-efficient of thermal expansion
θ	Dimensionless temperature, $= {}^{T-T_c}/_{T_H-T_c}$
θ_a	Atmospheric non dimensional temperature, $= {}^{T_a-T_c}/_{T_H-T_c}$
θ_w	Non dimensional temperature at the wall, $= {}^{T_w-T_c}/_{T_H-T_c}$
μ	Viscosity of fluid
ν	Kinematic viscosity of fluid
ρ	Density of fluid

Abstract

The study of a laminar natural convection in a square enclosure with conjugate boundary condition is done numerically. The top wall and right side wall of the enclosure are considered to have some finite thickness. The problem is solved using finite volume method and the multi-block method is used for meshing the domain. Various cases are considered by varying the parameters like characteristics Reynolds number, conductivity ratio and the wall thickness to analyse their effect on the heat transfer and flow characteristics. Reynolds number is taken equal to 10^3, 5×10^3 and 10^4 to constrain the flow as laminar in enclosure. The result shows some significant dependence on Reynolds number in the flow and temperature field inside the domain. An effect of insulation near the top wall is also studied and compared with the case of without insulation by varying the same parameters. It is to be noted that the average temperature inside the enclosure is reduced considerably due to the insulation.

Keywords: conjugate heat transfer, heat transfer in enclosure, laminar, natural convection, multi-block method

CHAPTER 1

Introduction

The study of the heat transfer in the enclosure has been prime topic for the researchers because of its wide range of application in the engineering and practical life. It finds its application from the cooling of nuclear reactor to the heat removal of the micro-electronic components and also for designing of the thermal conditioning of the room, thermal designing of the commercial buildings, cryogenic storage, furnace and many other. Besides its wide range of application, the simultaneous consideration of the conduction in wall and convection in fluid flow, remains an interesting field of investigation over the past decades.

Numerous articles could be found in the literature for studying the natural convection in the enclosure[1–16]. There are many boundary condition that could be considered but more complex and practically applicable domain are found to be less studied in literature. Most of the already studied domain are either bottom heating or side heating. But in this study we have considered the practically applicable boundary condition along with conjugate heat transfer.

1.1 Buildings and Energy

Energy has always been an important topic of discussion amongst the researchers. During any development program of any technology energy consumption has always been given significant importance. As accounted by Cole and Kernan [17] and Ramesh et al [18] a major part of the total energy consumed in a life time of any building is operating energy. This involves the maintenance and also most importantly the energy involved in maintaining the building in comforting condition both thermal and visual. The building energy analysis tool used prior to start of any project aims to reduce this operating energy.

It is important to understand that this reduction of energy consumption should be achieved while not at the cost of the reduction in performance. So the building must provide comfortable environment as compared to its outside environment. Earlier studies have proved the fact that the occupant quickly responds to any discomfort to regain their comfort, however this may adversely affect the energy consumption. Therefore precise prediction of thermal comfort is very important while designing the building to maintain lower energy consumption as stated by Nicol[19].

1.2 Review of Literature

The chronicle of analytical consideration and the recognition of the importance of studying the convection heat transfer in the enclosure goes back to the 1954 [3]. Batchelor had foreseen its application in building thermally efficient rooms before studying this problem. But the recent development of nuclear reactor, electronic component and many such equipment have increase the need for their in depth study. After Batchelor's work many researchers have explored this field. The work of Davis [6, 20] has been followed for many numerical investigations. Many of this simple problem is further studied for observing the effects of the various factors such as the inclined domain as done by Kuyper et al.[11] and Aydin et al [2], the effect of heat source inside the domain was studied by Frederick and Berbakow, Kuznetsov and Sheremet[8, 12], whilst the investigation on turbulent natural convection was done by Kuznetsov and Sheremet [14] and experiment was done by Betts and Bokhari [21] and the effect of radiation along with other modes of heat transfer was studied by Balaji and Venkateshan[4], Akiyama and Chong[1] and Ayachi et al. [22].

For in-depth understanding of this problem the comprehensive experiment was carried out and studied by Yin et al. [23] and Kim and Viskanta [24] with conducting wall. Kim and Viskanta [24] considered the square domain whilst Yin et al. [23] considered the tall rectangular cavity to study the heat transfer phenomenon. The latter experiment was designed for studying the effect of large aspect ratio. The low Reynolds number turbulence study of this natural convection in square cavity was studied by Henkes et al. [10].

Several in depth, up to-date discussion of heat transfer analysis inside the enclosure could be found in the literature related to convective heat transfer [9] and [25].

All of the above mentioned studies did not consider the effect of the wall thickness in their heat transfer analysis, albeit in actual practice we can not ignore it either. Kaminski and Prakash [26] considered the effect of conjugate heat transfer by considering one of the wall with finite thickness. In their study they considered steady, laminar and natural convection flow in square enclosure numerically. To solve the problem numerically, they used the method suggested by S. V. Patankar [27]. The basic idea behind solving problem by this method is to implicitly implement the no slip boundary condition in the solid region along with solving the complete flow field inside the fluid region. The above condition is obtained by setting a very large value of viscosity for the solid domain. This method is readily accepted by various researchers and is frequently followed to solve the various problem of the convection in the enclosure. Stream line and isotherms plots were given in the results for various values of Grashof number. They have also studied the variation of conductivity ratio and thickness ratio by treating the combination as one parameter and concluded that the results mainly depend on the product of these two ratios. Almost similar study was carried out by Misra and Sarkar [15]. They studied problem of conjugate heat transfer for the enclosure and solved by the same concept of S.V Patankar of assuming higher viscosity in the solid zone. The results are given for the Rayleigh number in the range of 10^3 to 10^6. The effect on Nusselt number and dimensionless temperature at the interface were studied for the wide range of the conductivities and the wall thickness. They have discussed the convergence characteristics of the equations by the application of the increasing viscosity on the solid zone and also the streamlines were shown for the various viscosity assumptions in the solid zone. The domain was simple as they have considered only one of the sides with its thickness and also same side is cold one while the side opposite to it was considered to be hot. The other sides were assumed to be thermally insulated.

Du and Bilgen [7] studied the effect of coupling of the conduction in solid wall and convection in fluid flow for the various parameters such as the conductivity ratio, aspect ratio, Rayleigh number and solid wall thickness. They considered the simple two dimensional domain with the constant heat flux applied to the solid vertical wall and the opposite wall is assumed to be insulated. The remaining two horizontal wall were considered to be at lower temperature. Their results stated that for the thinner wall the input parameters of low Rayleigh number, high aspect ratio and high conductivity ratio causes the heat transfer by the conduction to be more dominant.

This work of the natural convection for a cavity could be studied and applied for the specific practical problem such as room cooling as did by Horikiri et al. [28, 29]. They have studied the natural and forced convection heat transfer for a room with a heat source and wall with some finite thickness for ventilated 2D domain and then extended their study for the ventilated 3D domain with heat source. They have provided interrelation between the heat source arrangement, the effect of wall thickness, the evaluation of the thermal comfort level and also the energy consumption.

A simple two dimensional rectangular enclosure was studied recently by Kuznetsov and Sheremet [12–14] and the numerical study of the conjugate natural convection with a heat source of constant heat transfer rate with its convection radiation heat exchange with its one of the boundary was carried by considering the enclosure of some finite thickness. Cooling of electronic component finds application of the domain considered in there study. The time dependent study for studying the effect of the Grashof number, thermal conductivity of solid wall and the vertical position of the heat source was done.

Various techniques have already been proposed to solve the problem of the conjugate heat transfer. This involves the multi-domain method of Zhang et al. [30], immersed boundary method by Nagendra et al. [31], ghost method by Carlson et al. [32] or the oldest of them all, frequently followed and often used method suggested by S.V. Patankar[27]. The method followed in this study is different from the above all. We have used multi-block method to solve all the equations. Although it appears similar to the Wei Zhang's multi domain method it is slightly different from it. We have divided the whole domain into number of blocks, 3 for the case of without insulation and 5 for the case of with insulation. The whole fluid region is ascertained as a single block and two walls being considered as the either blocks. Whilst, for the case of insulation, the insulation forms one of the block and due to provision of insulation just below the top wall, the fluid region is divided in two, forming two of those blocks along with the blocks of solid walls. Thus, not only a new method is adopted to solve the problem but also the consideration of the conjugate wall with the constrained boundary condition is found to be practically applicable in this engineering world. The advantage of using this method is that it simplifies the solution method giving more accurate results. Also the boundary condition at the interface is implicitly treated.

In present study, investigation of laminar natural convection inside the enclosure is carried out for various sets of Reynolds number, conductivity ratio of solid wall and thickness of the solid wall. The enclosure is considered to be a square domain with isothermal left and top wall while the bottom wall is assumed to be perfectly adiabatic. Left wall is relatively cooler than the top and right wall. Top wall is assumed to be at highest temperature in the whole domain. The right wall is having convective heat transfer with the atmospheric air. Also the top wall and the right wall is considered to have some finite thickness which imposes the conjugate nature of problem. The effect of insulation considered near at the top wall of the enclosure is also studied for the same parameters as mentioned above. The comparison of results for both of the cases is made and is discussed.

1.3 Definition of the problem

When we talk about the enclosure here it covers a very broad area from a warehouse to small cold storage room where a heat is to be preserved. The current study was motivated from the fact that although most of the researchers are working to improve the efficiency of the cooling devices, very few have taken interest to develop thermally optimised enclosure or environment numerically which can reduce the load of such devices and thus reducing the energy consumption. The non-dimensionalisation allows us to apply the obtained result to any scale and depends only on the non dimensional number. Also the consideration of conjugate heat transfer allows us to understand its effect and gives us physical resemblance of the actual system. The effect of the different parameters, with and without insulation, are studied in the present work. There comparison is undertaken in present study and result gives us the insight the clear idea about the parametric range to be considered while designing any small cold storage or a commercial building.

1.4 Objectives and methodology

The main objective of this work was to study the effect of the various parameters on the heat transfer and flow characteristics of the air trapped inside of the enclosure. The CFD model is developed for this problem and a non-dimensionalised study was carried out so that the model could be applied for various scale. The finding of the current work will also present a reference for further developing of CFD model for enclosure with conjugate heat transfer.

To achieve these objectives the work was conducted in following stages:-

- **Stage 1. Literature survey :-** Investigation of the already published work in various SCI journals on CFD modeling of enclosure and various methods and computational domains already studied was done.

- **Stage 2. CFD Model :-** Solving governing equations, non-dimensionalisation of this equations and developing suitable CFD model for the selected computational domain was done.

- **Stage 3. Code Validation :-** Comparison of the numerical results obtained with the current code with already published work in SCI journals was carried out.

- **Stage 4. Model Assessment :-** A detail parametric study for understanding the effect of boundary condition on the selected computational domain was done.

- **Stage 5. Result Analysis :-** The result obtained was analysed for better understanding of the various parameters with the help of plots and graphs.

CHAPTER 2

Mathematical Formulation

Initially a simple square enclosure is considered filled with air with given boundary conditions. Then an insulation is provided just below the top wall to study its effect. All the four boundaries are having different and practical boundary conditions as shown in figures 2.1 . The two of the walls, top wall and the right wall, are assumed to have some finite thickness to impose the conjugate boundary conditions. The bottom wall is assumed to be thermally insulated whilst the top wall is assumed to have constant hot temperature. The left wall is maintained isothermally at lower temperature and the right wall is having convective interaction with air at environmental conditions. Three different cases were considered by varying the thickness of both of the boundary walls simultaneously and for each case the Reynolds number and the conductivity of the solid wall are varied to study the effect of these two parameters on the flow and temperature field of fluid in enclosure. The insulation is provided at the height of 0.9 of the length of the side and the thickness of the insulation is as small as 0.01.

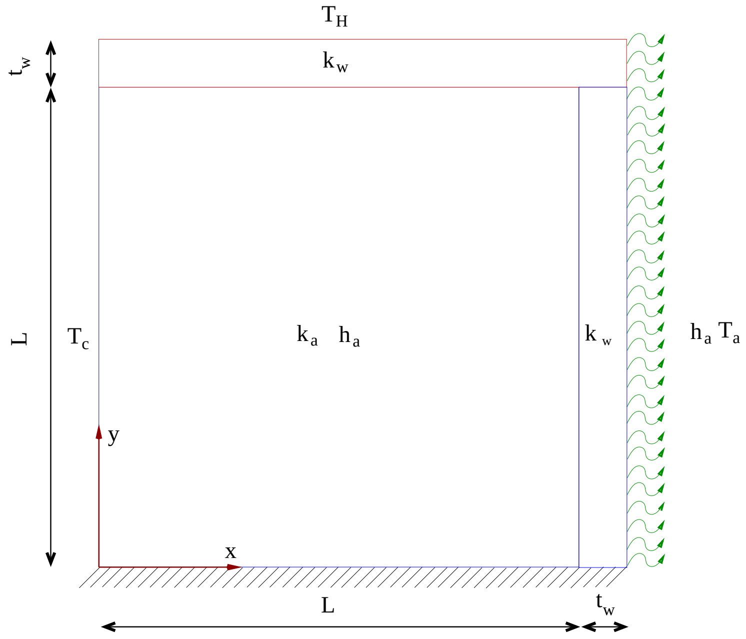


Figure 2.1: Boundary conditions of domain

2.1 Governing differential equation

The governing equations for two dimensional, steady state flow are presented as

continuity equation,

$$\frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) = 0$$

u-momentum equation,

$$\frac{\partial(\rho u u)}{\partial x} + \frac{\partial(\rho v u)}{\partial y} = \frac{\partial}{\partial x}\left(\mu\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu\frac{\partial u}{\partial y}\right) - \frac{\partial p}{\partial x}$$

v-momentum, equation,

$$\frac{\partial(\rho v u)}{\partial x} + \frac{\partial(\rho v v)}{\partial y} = \frac{\partial}{\partial x}\left(\mu\frac{\partial v}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu\frac{\partial v}{\partial y}\right) - \frac{\partial p}{\partial y} + \rho g \beta (T - T_C)$$

energy equation,

$$\rho c_p\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right) = \frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)$$

2.1.1 Assumptions

The domain is assumed to be two dimensional with laminar natural convection mode of heat transfer. The velocity components along the x and y directions are u and v respectively. All the properties of the fluid in the enclosure are assumed to be constant. And, the density is taken as a constant except the buoyancy term which is governed by the Boussinesq approximation. Also, heat transfer by radiation is assumed to be negligible as compared to the other modes of heat transfer and so radiation terms are neglected. The conductivity of the insulation material is taken same as that of the air.
The above equations are non diamensionalised by following dimensionless parameter,

$$
\begin{array}{llll}
U=\dfrac{u}{U_0}, & V=\dfrac{v}{U_0}, & X=\dfrac{x}{L}, & Y=\dfrac{y}{L} \\
K_r=\dfrac{k_s}{k_f}, & P=\dfrac{p}{\rho U_0^2}, & \theta=\dfrac{T-T_C}{T_H-T_C}, & \mathrm{Pr}=\dfrac{\nu}{\alpha} \\
K^*=\dfrac{k}{k_f}, & \rho^*=\dfrac{\rho}{\rho_f}, & c_p^*=\dfrac{c_p}{c_{pf}} &
\end{array}
$$

The x and y are normalised with respect to the maximum length of the domain L and the velocity is scaled by U_0, obtained by setting the Richardson number as unity, where $U_0 = \sqrt{g\beta l(T_H - T_C)}$.
The non-dimensionalised governing equations can be given as :
Continuity Equation:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0$$

X-momentum equation:

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\left(\frac{1}{Re}\right)\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right)$$

Y-momentum equation:

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial X}+\theta+\left(\frac{1}{Re}\right)\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)$$

Energy equation:

$$\rho^* c_p^* \left(U\frac{\partial\theta}{\partial X} + V\frac{\partial\theta}{\partial Y} \right) = \left(\frac{K^*}{Re.Pr} \right) \left(\frac{\partial^2\theta}{\partial X^2} + \frac{\partial^2\theta}{\partial Y^2} \right)$$

It should be noted here that the momentum equations are solved only for the fluid region.

2.1.2 Boundary conditions

All the four boundaries of the square domain are maintained at the different boundary conditions. The left boundary is maintained at the lower temperature, i.e. cold wall. The top wall is considered to be maintained at the high temperature in the whole domain. A convective heat loss is considered through the right wall which is also considered having some finite thickness. The bottom of the enclosure is assumed to be perfectly insulated. The figures 2.1 show the boundary conditions considered in the case of with insulation and without insulation respectively.

The u and v are assumed as zero along the boundary wall due to the no slip condition. Thus, the relevant boundary conditions are given as

$\frac{\partial\theta}{\partial Y} = 0$ and u=v=0 for $Y = 0$ and $0 \leqslant X \leqslant X_{max}$.....insulated boundary

$\theta = 0$ and u=v=0 for $X = 0$ and $0 \leqslant Y \leqslant Y_{max}$ .....isothermal cold wall

$\theta = 1$ and u=v=0 for $Y = Y_{max}$ and $0 \leqslant X \leqslant X_{max}$ ......isothermal hot wall

$K^* \frac{\partial\theta}{\partial X} = B_i(\theta_a - \theta_w)$ for $X = X_{max}$ and $0 \leqslant Y \leqslant Y_{max}$..... convective heat transfer

X_{max} and Y_{max} vary according to the thickness of the solid wall of the domain.

2.2 Numerical Analysis

The method which is most commonly followed in the past is based on the Patankar's [27] method. This method is based on the assumption of the high viscosity value for the solid region when conjugate heat transfer is to be considered. So, the whole domain can be considered as one and with such assumption the velocity term can be neglected in the solid zone and simultaneous solution of the equation is possible. Thus, the energy equation is solved for both the zones but the momentum equation was solved for fluid region. And few other methods developed in the recent past involves the immersed boundary condition and ghost node method.

2.2.1 Method of solution

The method adopted here to solve the problem is known as the multiblock method. This application of the method for the conjugate boundary problem is not seen in the literature before. The previously followed method were based on the assumptions of S. V. Patankar. But the method adopted here has no such need of assumptions. In multiblock method, the domain is divided in the suitable number of blocks separating the fluid region from the solid region. This allows one to apply the respective equation to the respective blocks, i.e the momentum equation only for the fluid and energy equation for all of the regions.

The diffusive term is discretised with central difference scheme and the variables are stored according to the collocated arrangement. The governing equations are solved using finite volume approach. The semi-implicit method for pressure linked equation (SIMPLE) is used to couple momentum and continuity equations.

2.3 Grid In-dependency Test

The non-uniform grid is used for discretising the computational domain. The domain is divided into the blocks and gridding of each individual block is carried out. There are three blocks in total where block 1 is fluid zone and remaining two blocks are solid zones. The grids are kept finer near the wall, where more disturbances are expected to happen and coarser at the center for all of the blocks. Figure 2.2 shows the grid used for carrying out the computations. A multiblock grid system having three blocks of 200×200, 40×200, and 240×40 was found to be sufficient to resolve the details of flow and temperature fields inside the domain.

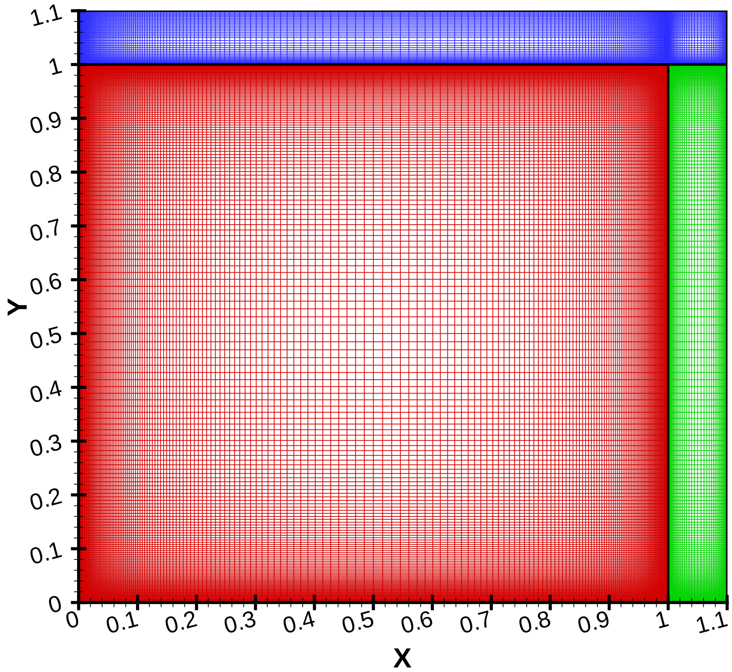
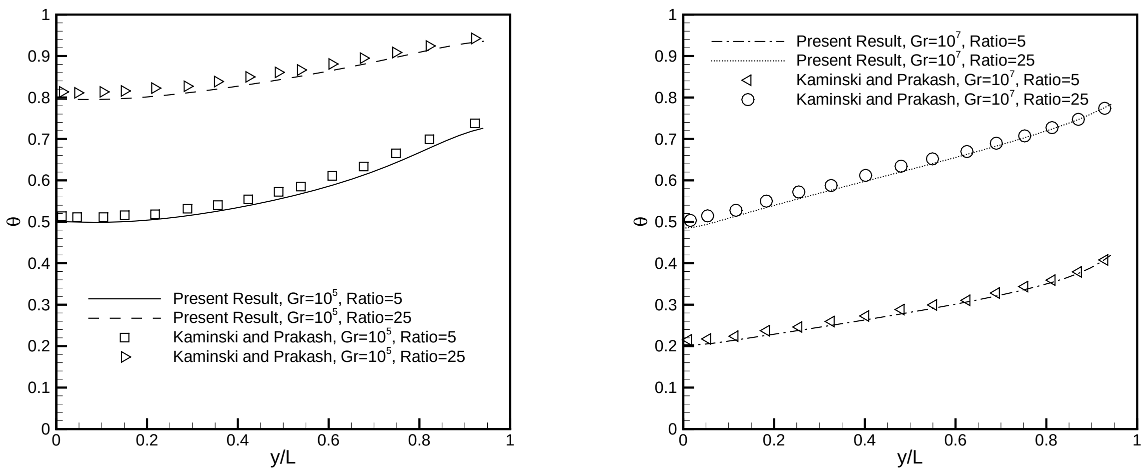


Figure 2.2: Gridding of multi block domain

2.4 Validation of the Computer Code

For the purpose of code validation, the problem of Kaminski and Prakash [26] is considered. They considered natural convection problem in a two dimensional conjugate enclosure with a solid vertical wall for $Gr = 10^5$and $10^7$for the ratio of 5 and 25 respectively, where ratio corresponds to $\frac{k_w L}{kt}$. The calculated non-dimensional temperature $\theta = \frac{T - T_C}{T_H - T_C}$ at the interface for the current problem are compared with the results obtained by Kaminski and Prakash [26]. The predicted interface temperature agrees quite well with the published results by Kaminski and Prakash as shown in figure 2.3.



(a) $Gr = 10^5$ (b) $Gr = 10^7$

Figure 2.3: Validation of the code

CHAPTER 3

Results and discussions

Two dimensional laminar natural convection of an air filled square enclosure is studied for different parameters such as characteristics Reynolds number, conductivity ratio and thickness of the wall. Reynolds number is varied between 10^3 and 10^4 for various conduction ratio in the range of $10 \leqslant K_r \leqslant 100$ in 9 equal steps as shown in table 3.1 . To study the effect of wall thickness on the heat transfer characteristics, it is varied as 0.05, 0.075 and 0.1 times the enclosure dimension. The isothermal contours, stream line contours and velocity vectors are presented. The thermal conductivity of fluid, i.e. air is considered to be 0.0239 W/m-K.

The results show some interesting facts and phenomenon which were unexpected at the early stage of our investigation. Simulations for various parameters were carried with the same computer program and effects of various parameters on the result are presented. They are shown in tabular form in the following part and an effort to correlate them are made . Reynolds number is considered so as to have the laminar flow inside

k_s	0.25	0.5	0.75	1.00	1.25	1.50	1.75	2.00	2.39
K_r	10.417	20.83	31.25	41.67	52.08	62.5	72.917	83.33	100

Table 3.1: Conductivity ratio

the enclosure. Another parameter which could not be neglected for the problem specified here is conductivity of the solid wall. Thus, the effect of the solid wall conductivity is also studied and it is varied from 0.25 to 2 W/m-K. Also the effect of the insulation provided at the top is compared with the results of the simulation of the enclosure without insulation.

3.1 Heat transfer and fluid flow characteristics of square enclosure without insulation

A 2-D square enclosure is considered with the wall of finite thickness. This imposes the conjugate nature of heat transfer. The effect of the boundary condition which causes the buoyancy driven flow inside the enclosure. Thus a laminar natural flow is induced inside the enclosure. In the following section the a detail parametric study is carried out discussing the effect of this various parameters.

3.1.1 Flow Characteristics

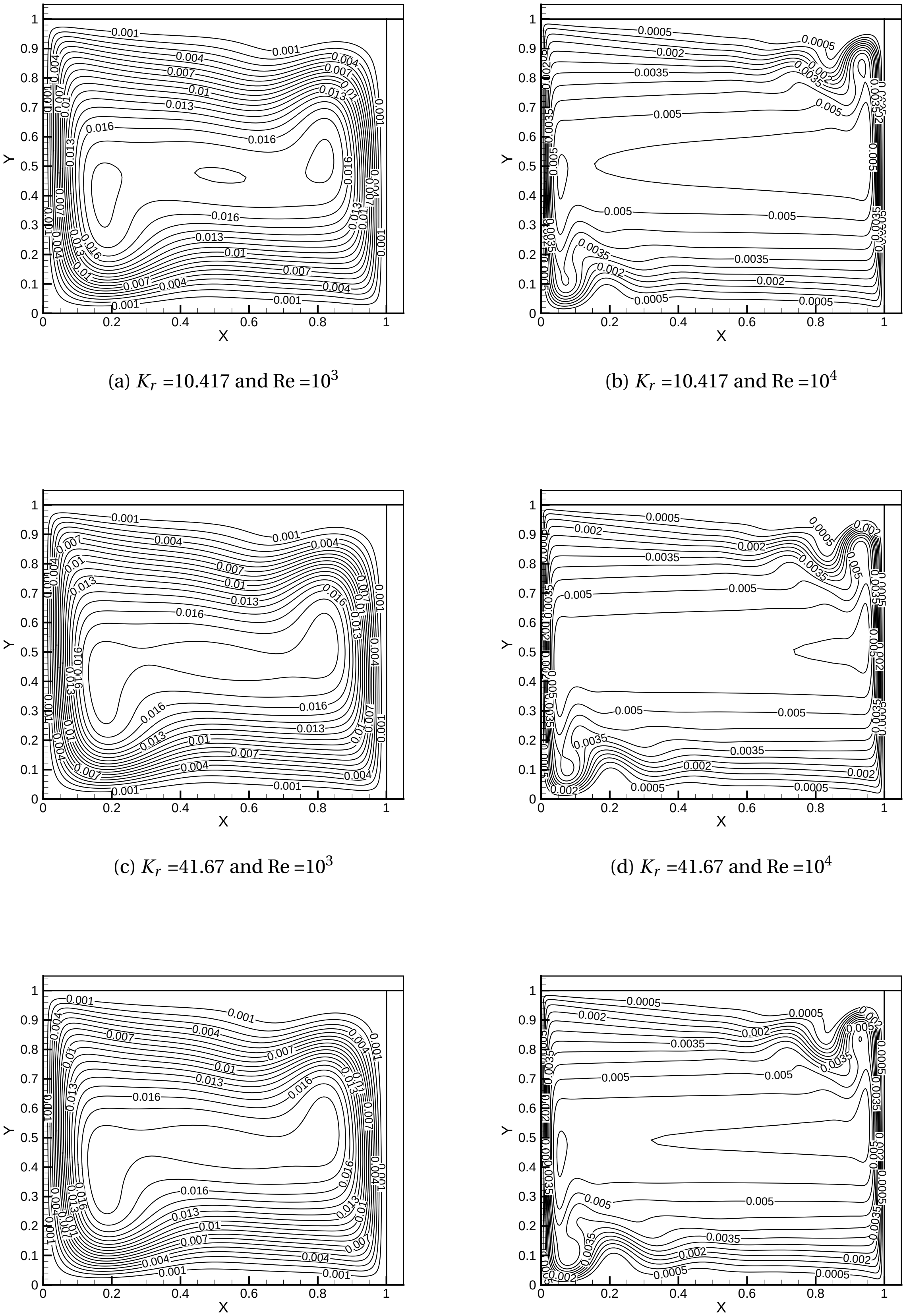
Figures 3.1 and 3.3 show the streamlines for the thickness of the wall as 0.05 and 0.1 respectively. The left side of the figure represents the streamlines for a lower Reynolds number of 10^3 whilst the right side is for the higher Reynolds number of 10^4. The figure also presents the effect of conductivity ratio, K_r. It is noticed that the strength of convection increases with the increase in conductivity ratio, K_r. Being a natural convection, temperature difference drives the flow within the enclosure and as the conductivity of the wall increases, more heat leaks inside the cavity which results in an increase in the strength of convection. The streamline value is observed to be on the higher side for the lower Reynolds number compared to the higher Reynolds number as it should be. The stream lines near the vertical wall are observed to be more dense which indicates the high velocity and also thin boundary layer for higher Reynolds number for all the cases. Some perturbations and disturbances could be observed in the streamline at the lower left corner and the upper right corner for the case of higher Reynolds number.

The velocity vectors shown in figure 3.4 are observed to change according to the Reynolds number but its variation with conductivity ratio and the wall thickness is less. The velocity vectors for the lower Reynold number is observed to be dispersed inside the cavity but as the Reynolds number increases the vectors are observed to be more denser near the wall as it should be. Also, it is to be noted that the fluid bounces back off the horizontal wall. This could be explained as: the fluid flowing along the wall strikes the horizontal wall and the fluid bounces back due to the inertia of the moving fluid. This is seen at the bottom left and top right corner of the enclosure. Small vortices are observed in the streamline plots due to the same reason. The central area or the core part of the enclosure is almost stagnant which indicates there is very little or negligible flow.

3.1.2 Heat Transfer Characteristics

Isothermal plots shown in figures 3.5 and 3.7 show the variation of temperature field inside the enclosure for various Reynolds numbers, thicknesses and conductivity ratios.

The variation of temperature in side wall is more noticeable only for the combination of lower conductivity ratio and higher Reynolds number for both of the thicknesses as seen in figures 3.5b and 3.7b. This temperature distribution changes with increasing conductivity ratio and almost a constant temperature is observed in the side wall. Another noticeable fact is that the average temperature of the side wall is decreasing with the increase in the Reynolds number, signifying the lower temperature zone or more cooler area inside the enclosure. The atmospheric temperature is assumed to be 0.7 on non-dimensional scale, this explains the perpendicular intersection of the isotherm line of 0.7 with right side wall.



(a) K_r =10.417 and Re =10^3

(b) K_r =10.417 and Re =10^4

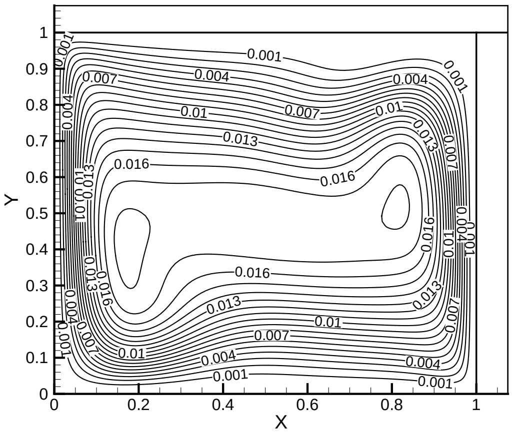
(c) K_r =41.67 and Re =10^3

(d) K_r =41.67 and Re =10^4

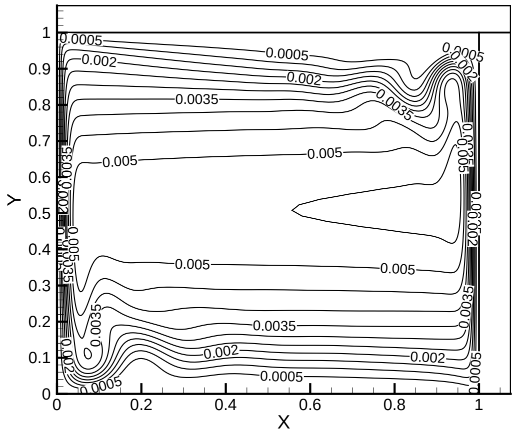
(e) K_r =100 and Re =10^3

(f) K_r =100 and Re =10^4

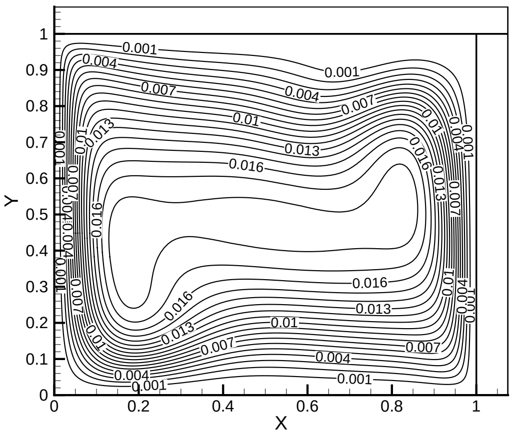
Figure 3.1: streamline plots for t=0.05



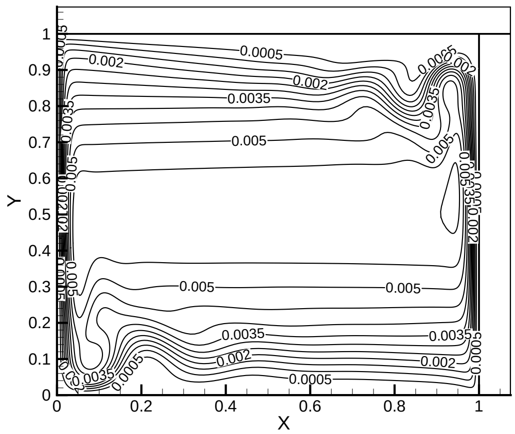
(a) K_r =10.417 and Re =10^3



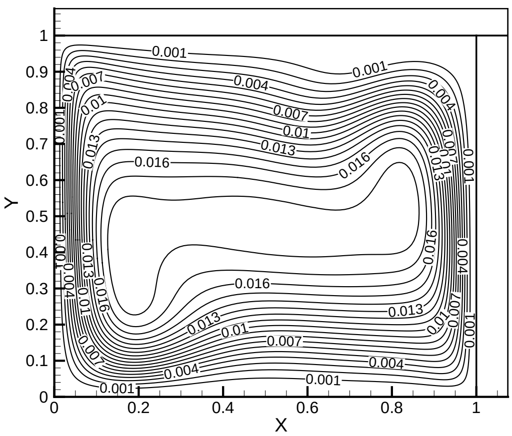
(b) K_r =10.417 and Re =10^4



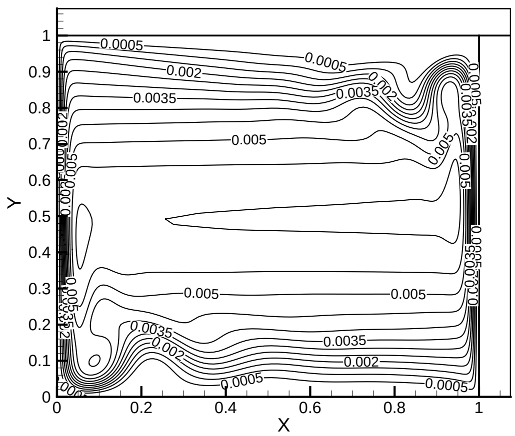
(c) K_r =41.67 and Re =10^3



(d) K_r =41.67 and Re =10^4

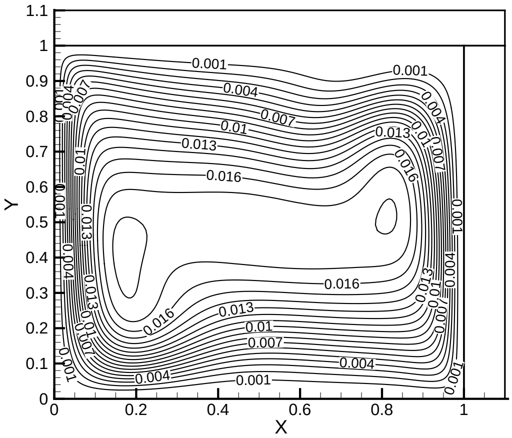


(e) K_r =100 and Re =10^3

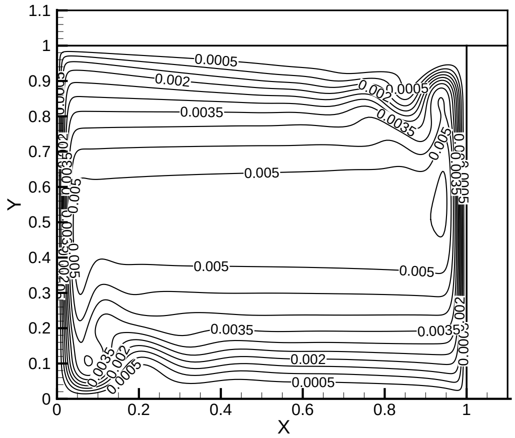


(f) K_r =100 and Re =10^4

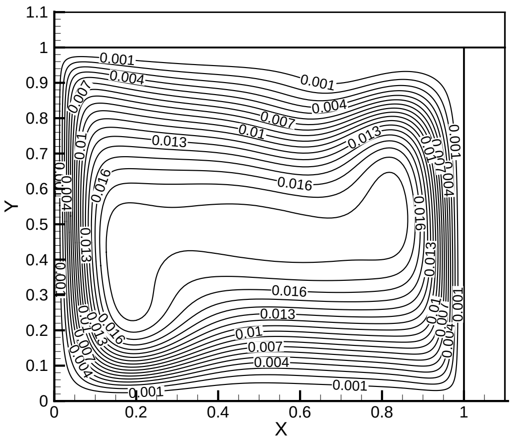
Figure 3.2: streamline plots for t=0.075



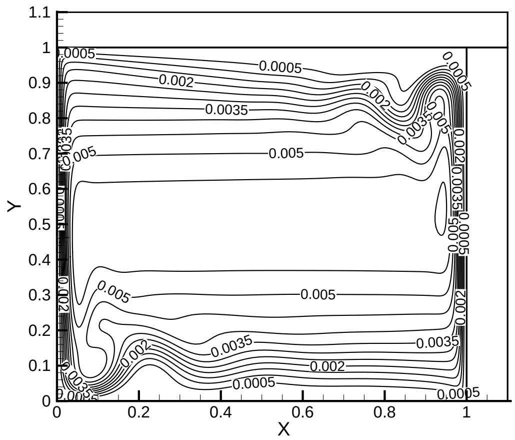
(a) K_r =10.417 and Re =10^3



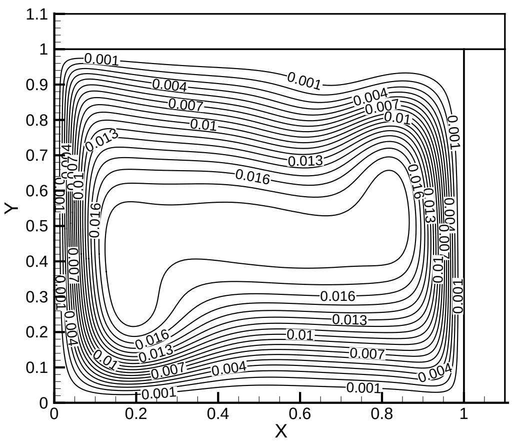
(b) K_r =10.417 and Re =10^4



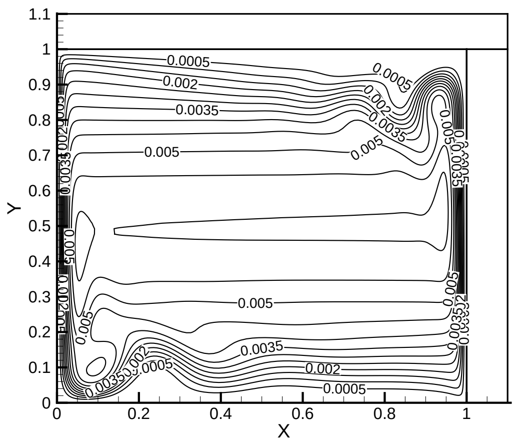
(c) K_r =41.67 and Re =10^3



(d) K_r =10.417 and Re =10^4

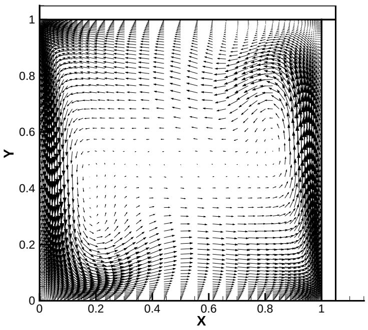


(e) K_r =100 and Re =10^3

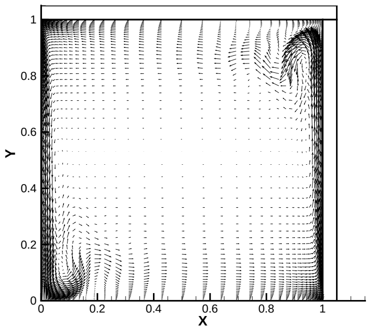


(f) K_r =100 and Re =10^4

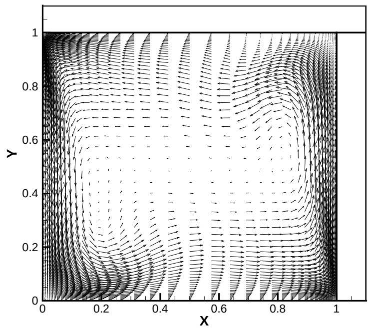
Figure 3.3: streamline plots for t=0.1



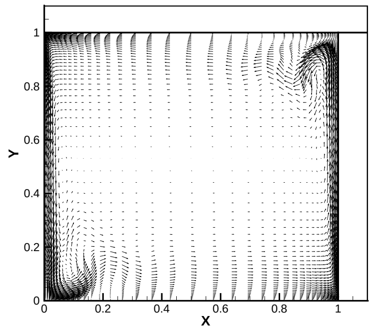
(a) t =0.05, K_r =10.417 and Re =10^3



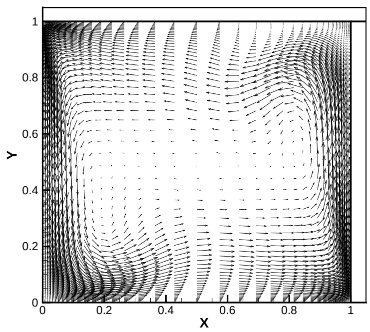
(b) t =0.05, K_r =10.417 and Re =10^4



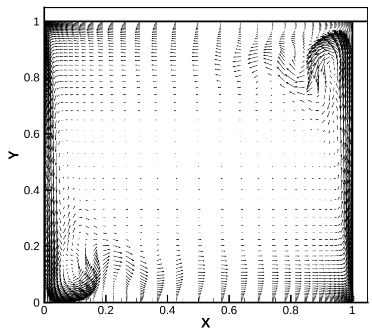
(c) t =0.1, K_r =10.417 and Re =10^3



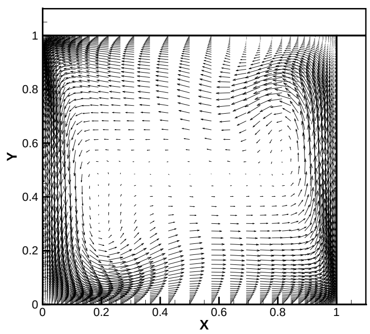
(d) t =0.1, K_r =10.417 and Re =10^4



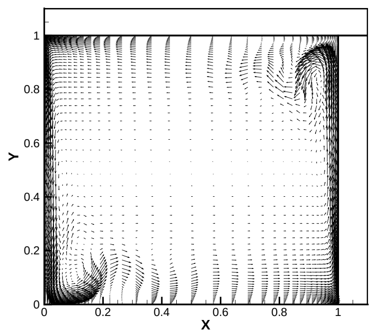
(e) t =0.05, K_r =100 and Re =10^3



(f) t =0.05, K_r =100 and Re =10^4

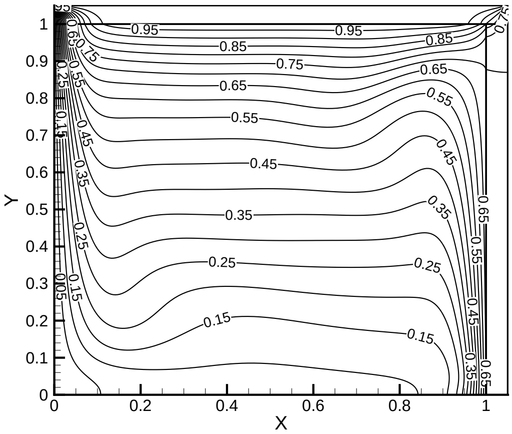


(g) t =0.1, K_r =100 and Re =10^3

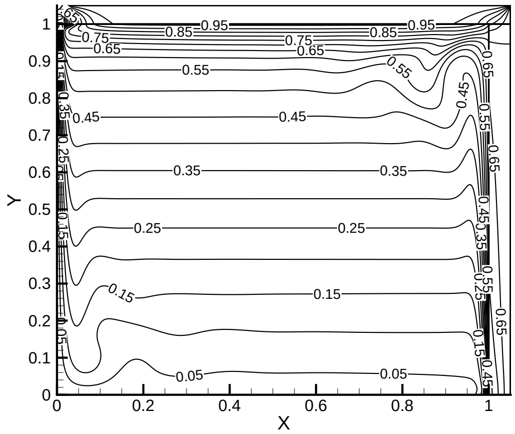


(h) t =0.1, K_r =100 and Re =10^4

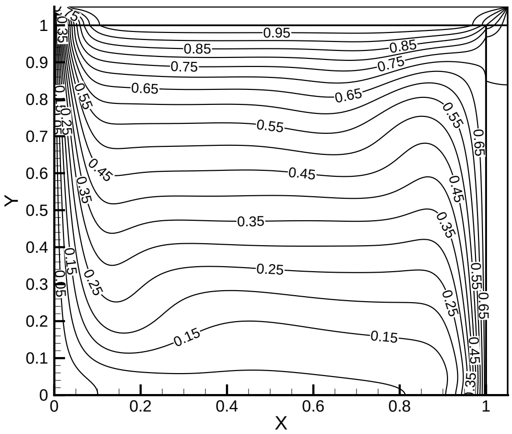
Figure 3.4: vector plots



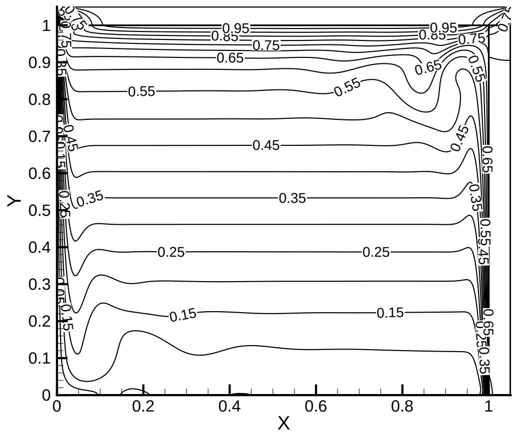
(a) K_r =10.417 and Re =10^3



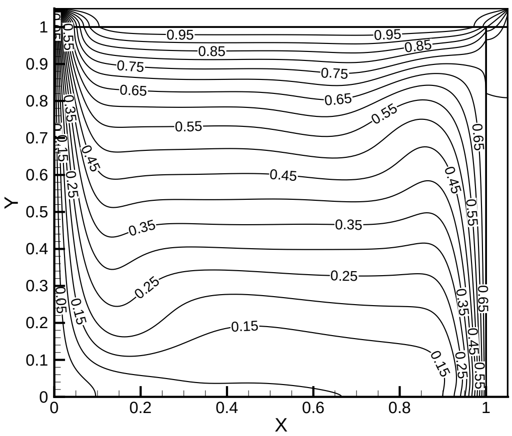
(b) K_r =10.417 and Re =10^4



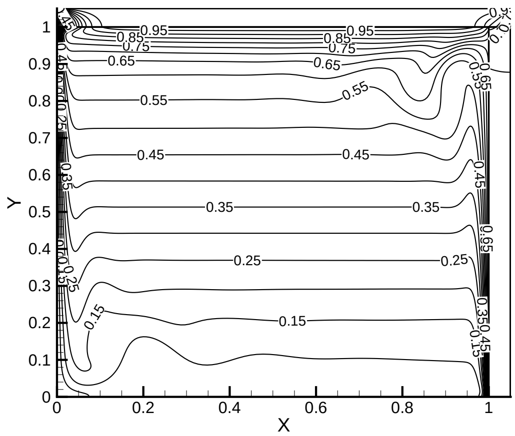
(c) K_r =41.67 and Re =10^3



(d) K_r =41.67 and Re =10^4

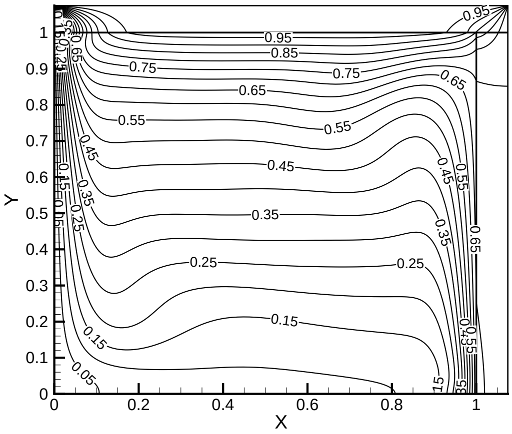


(e) K_r =100 and Re =10^3

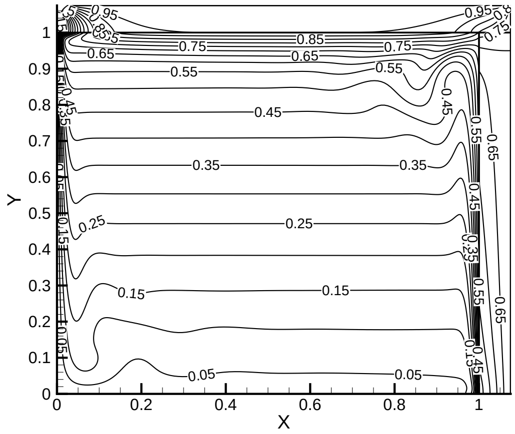


(f) K_r =100 and Re =10^4

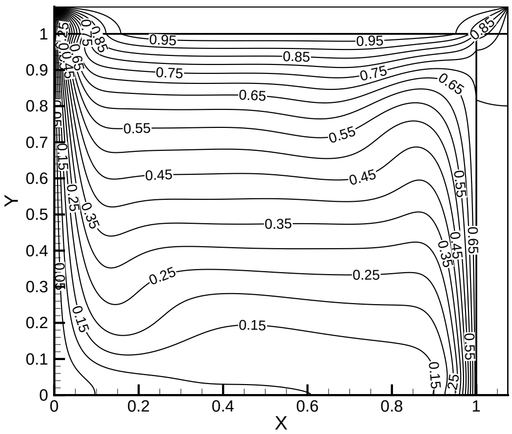
Figure 3.5: isotherm plots for t=0.05



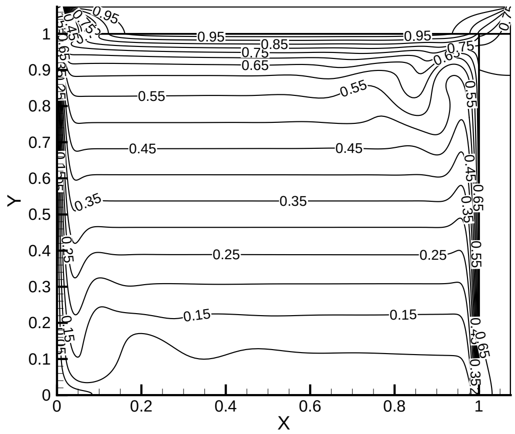
(a) K_r =10.417 and Re =10^3



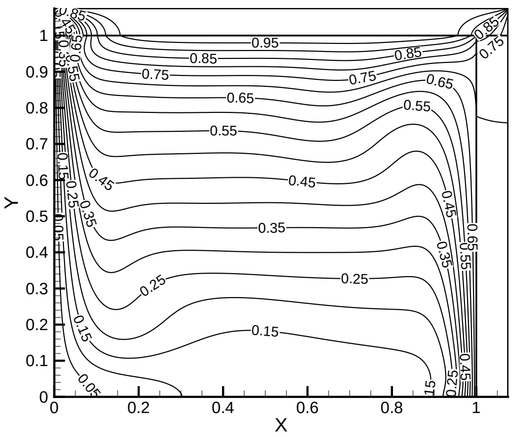
(b) K_r =10.417 and Re =10^4



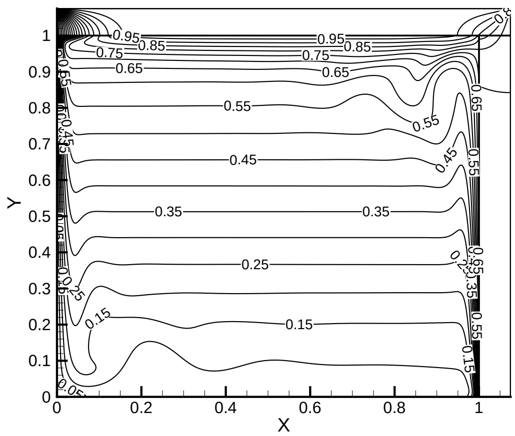
(c) K_r =41.67 and Re =10^3



(d) K_r =41.67 and Re =10^4

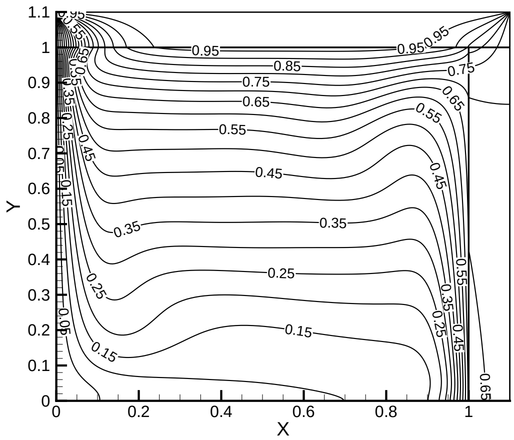


(e) K_r =100 and Re =10^3

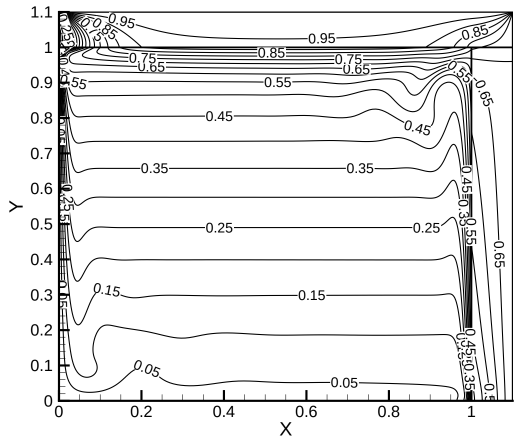


(f) K_r =100 and Re =10^4

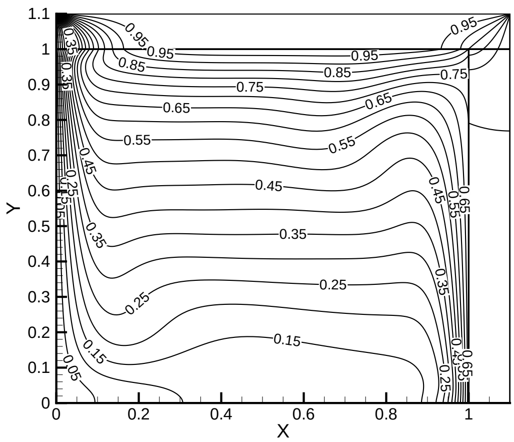
Figure 3.6: isotherm plots for t=0.075



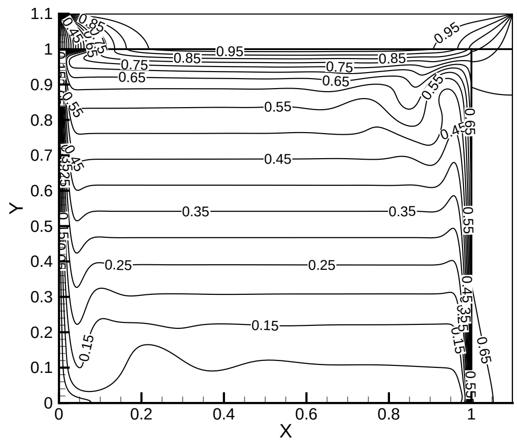
(a) K_r =10.417 and Re =10^3



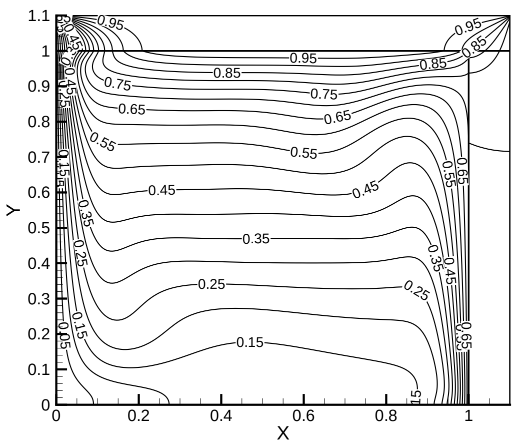
(b) K_r =10.417 and Re =10^4



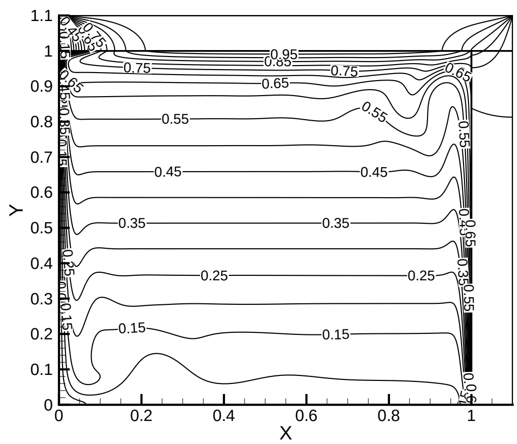
(c) K_r =41.67 and Re =10^3



(d) K_r =41.67 and Re =10^4



(e) K_r =100 and Re =10^3



(f) K_r =100 and Re =10^4

Figure 3.7: isotherm plots for t=0.1

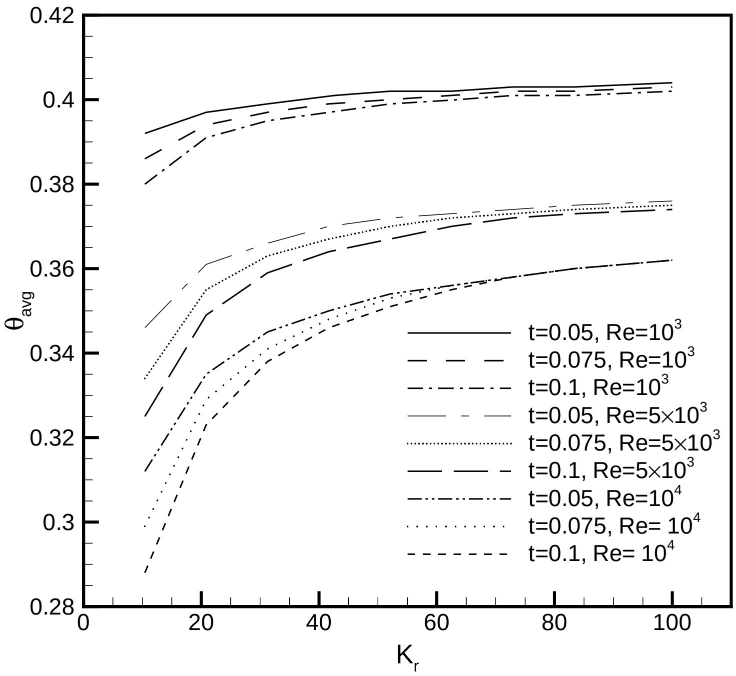
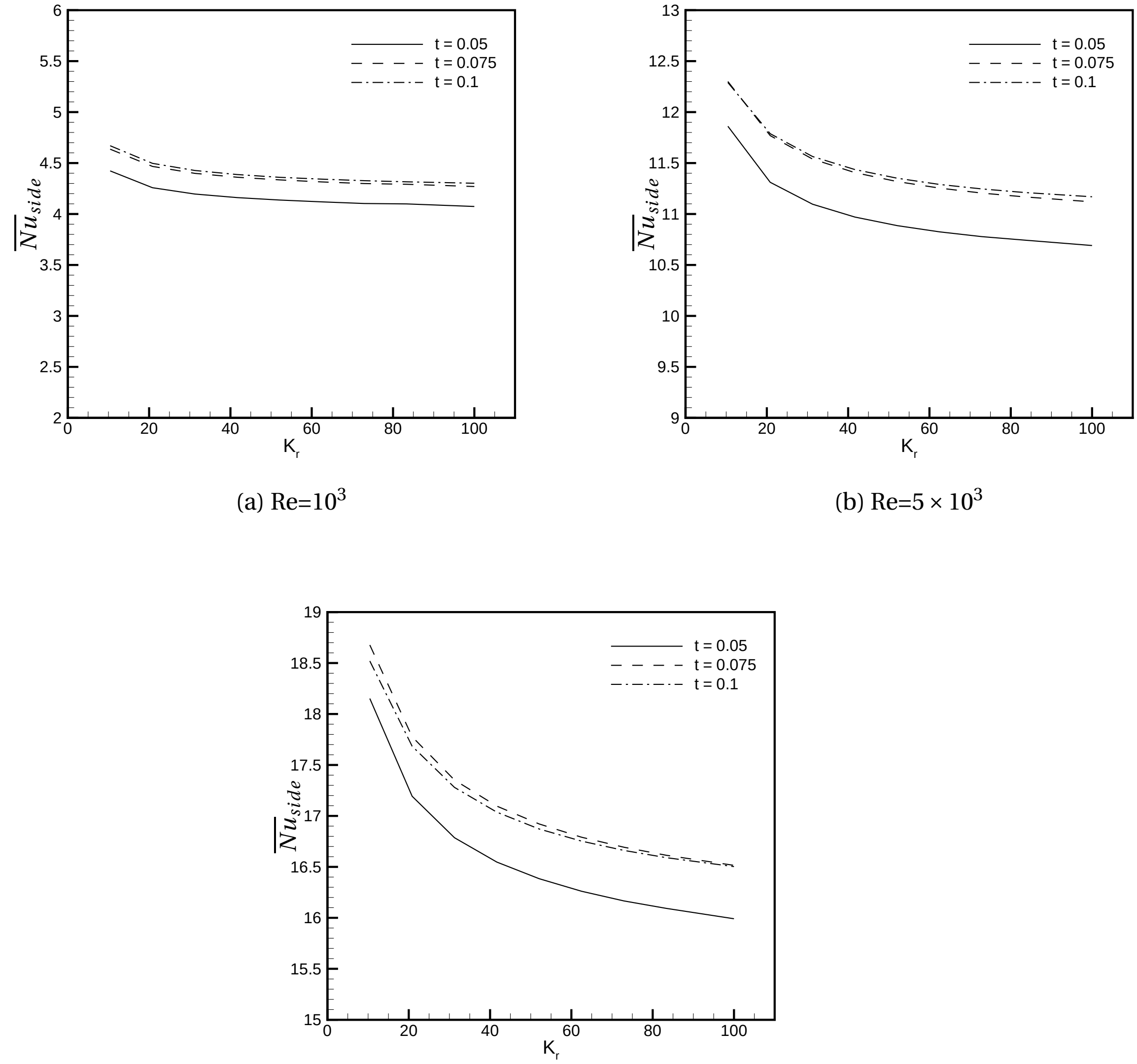


Figure 3.8: Average temperature for various cases

3.1.3 Average temperature in the enclosure

Conductivity of the wall has a significant role, but its effect on the average temperature is different from initial expectation. The plot of average temperature with the conductivity ratio for various thickness and Reynolds number is shown in figure. 3.8 . The higher average temperature is observed for the whole enclosure for lower Reynolds number as we observe in the figure. 3.8. But the temperature gradient for overall range of conductivity ratio is less when compared with higher Reynolds number. As the Reynolds number increases the temperature gradient also increases and slope becomes more steep. The results for the average temperature for the low Reynolds number are very much similar to each other and the variation being the same for all of the conductivity ratios. But, as the Reynolds number is increased from 10^3 to 5×10^3 and again to higher value of 10^4, the effect of the wall thickness on the average temperature is observed to be less significant at higher conductivity ratios. This shows that a desired temperature can be maintained in the enclosure irrespective of the wall thickness at higher Reynolds number.

The variation of average Nusselt number on the side wall ($\overline{Nu}_{side}$) over the whole range



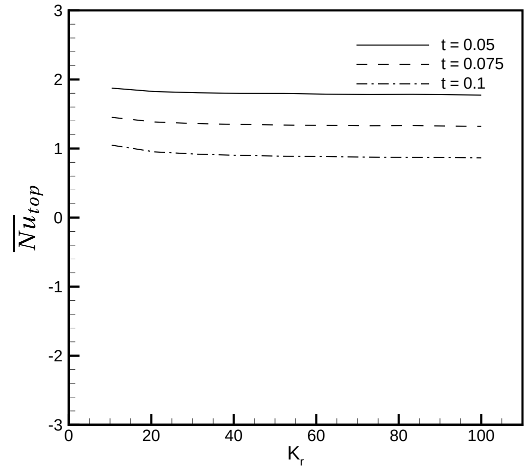
(a) Re=10^3

(b) Re=5×10^3

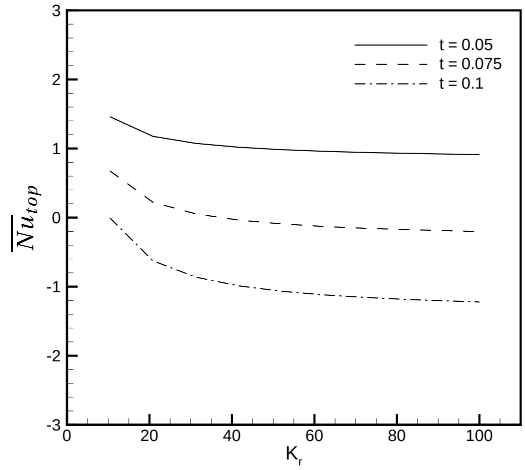
(c) Re=10^4

Figure 3.9: Average Nusselt number for side wall

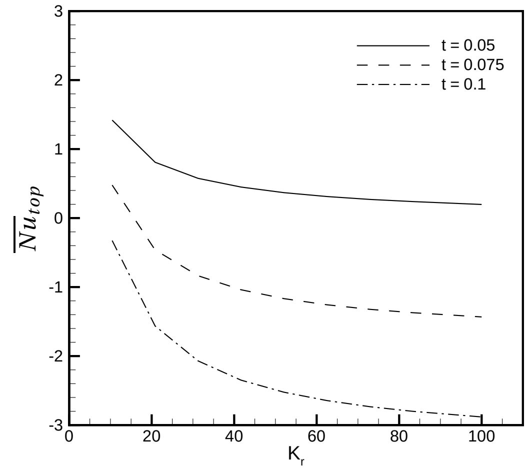
of conductivity ratios for various thicknesses and Reynolds numbers is shown in figure 3.9. A considerable change in the values of $\overline{Nu}_{side}$ can be observed for the various cases of Reynolds number. It should be pointed here that for a given wall thickness, there is a decrease in the value of $\overline{Nu}_{side}$ by almost 10% when conductivity ratio increases from 10.417 to 100, irrespective of the change in Reynolds number. On the other hand, $\overline{Nu}_{side}$ is found to increase with the increase in wall thickness for a given Reynolds number. This is because, with the increase in the wall thickness, the conductive heat transfer across the side wall decreases which results in an increase in the $\overline{Nu}_{side}$. Also, it should be



(a) Re=10^3



(b) Re=5×10^3



(c) Re=10^4

Figure 3.10: Average Nusselt number for top wall

noted that after a certain thickness of the wall there is no further increase in the average Nusselt number for the side wall. However, this effect is not observed for the top wall, where the average Nusselt number $\overline{Nu}_{top}$ keeps on changing with the change in wall thickness (see figure 3.10). For this case, the trend is entirely opposite; $\overline{Nu}_{top}$ now decreases with the increase in the wall thickness for a given Reynolds number. The variation in $\overline{Nu}_{top}$ is more pronounced at higher Reynolds number, suggesting a strong effect of convection on it. Also, $\overline{Nu}_{top}$ is found to decrease with the increase in Reynolds number as opposed to the case of side wall. The value of average Nusselt number decreases further with the increase in the conductivity ratio. Infact, at higher Reynolds number with high conductivity ratio, it becomes negative which suggests that heat leakage into the enclosure from the top is stopped.

3.1.4 Tabular form of the results

For a non dimensional wall thickness of $t = 0.05, t = 0.075$ and $t = 0.1$ the heat transfer from the side and top walls and the average Nusselt numbers are listed in Tables 3.2, 3.3 and 3.4 respectively.

The average value of various parameters in the enclosure, i.e. temperature, Nusselt number for side and top wall and heat transfer from side and top wall for the various cases are given in tabular form in Tables 3.2, 3.3 and 3.4. The variation for the Nusselt number for the side wall is more significant as compared to the top wall across the Reynolds number for all the cases. Also the change in Nusselt number across the top wall for the Reynolds number is observed to increases from 24% to 89% for higher K_r for $t = 0.05$. It is to be noted that the percentage change also increase with the thickness of the solid wall. In a nutshell the maximum variation is observed for the higher K_r and thicker wall. The Nusselt number for top wall is observed to be on the negative side for the enclosure with higher thickness. The values of the Nusselt number is in the range of the unity suggesting that the conduction and the convection contribute equally for the heat transfer across the top wall, whilst the range of the Nusselt number is more than unity for the side wall and therefore the convection is more dominant for the side wall.

Also the Nusselt number for the top wall decreases as the Reynolds number increases, as contrary to the case of side wall where the Nusselt number increases simultaneously with the Reynolds number. As shown in the figure 3.8 the average temperature is in the range of around 0.3 – 0.4. But, it is to be noted that though the average temperature is less for higher Reynolds number for all the cases but the variation of the temperature is also very sensitive for the variation of the K_r for higher Reynolds number as compared with the lower Reynolds number.

Table 3.2: Heat transfer and Nusselt number for wall thickness $t = 0.05$

$K_r \Downarrow$	Q_{side}			Q_{top}			$\overline{Nu}_{side}$			$\overline{Nu}_{top}$			θ_{avg}		
$Re \Rightarrow$	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4
10.417	2.932	7.383	10.758	1.7205	2.167	2.746	4.423	11.861	18.152	1.874	1.460	1.420	0.392	0.346	0.312
20.83	2.89	7.435	11.000	1.685	1.990	2.405	4.258	11.311	17.193	1.824	1.176	0.809	0.397	0.361	0.335
31.25	2.871	7.437	11.045	1.673	1.927	2.273	4.196	11.096	16.786	1.807	1.074	0.576	0.399	0.366	0.345
41.67	2.859	7.426	11.046	1.666	1.891	2.202	4.161	10.970	16.547	1.798	1.018	0.450	0.401	0.37	0.350
52.08	2.849	7.414	11.034	1.661	1.869	2.154	4.137	10.886	16.384	1.797	0.983	0.369	0.402	0.372	0.354
62.5	2.840	7.402	11.017	1.657	1.853	2.121	4.119	10.825	16.261	1.787	0.959	0.313	0.402	0.373	0.356
72.917	2.832	7.391	11.000	1.654	1.842	2.096	4.103	10.778	16.166	1.782	0.941	0.271	0.403	0.374	0.358
83.33	2.834	7.386	10.988	1.656	1.834	2.078	4.099	10.744	16.093	1.785	0.929	0.240	0.403	0.375	0.359
100	2.817	7.367	10.959	1.648	1.821	2.051	4.074	10.691	15.991	1.774	0.910	0.198	0.404	0.376	0.362

Table 3.3: Heat transfer and Nusselt number for wall thickness $t = 0.075$

$K_r \Downarrow$	Q_{side}			Q_{top}			$\overline{Nu}_{side}$			$\overline{Nu}_{top}$			θ_{avg}		
$Re \Rightarrow$	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4
10.417	3.012	7.306	10.372	1.507	1.924	2.495	4.636	12.3	18.677	1.451	0.675	0.478	0.386	0.334	0.299
20.83	3.000	7.538	10.957	1.466	1.704	2.074	4.467	11.769	17.777	1.384	0.222	-0.462	0.394	0.355	0.329
31.25	2.99	7.597	11.122	1.451	1.622	1.906	4.399	11.539	17.352	1.36	0.053	-0.836	0.397	0.363	0.341
41.67	2.981	7.618	11.186	1.444	1.579	1.815	4.361	11.406	17.096	1.348	-0.037	-1.039	0.399	0.367	0.348
52.08	2.974	7.626	11.214	1.439	1.552	1.757	4.335	11.318	16.921	1.34	-0.092	-1.168	0.4	0.37	0.353
62.5	2.968	7.628	11.224	1.435	1.534	1.717	4.315	11.254	16.791	1.334	-0.13	-1.257	0.401	0.372	0.356
72.917	2.962	7.627	11.228	1.431	1.521	1.687	4.299	11.205	16.693	1.329	-0.157	-1.323	0.402	0.373	0.358
83.33	2.961	7.628	11.230	1.433	1.512	1.665	4.292	11.168	16.614	1.331	-0.176	1.372	0.402	0.374	0.36
100	2.95	7.622	11.224	1.426	1.499	1.636	4.27	11.119	16.515	1.321	-0.203	-1.433	0.403	0.375	0.362

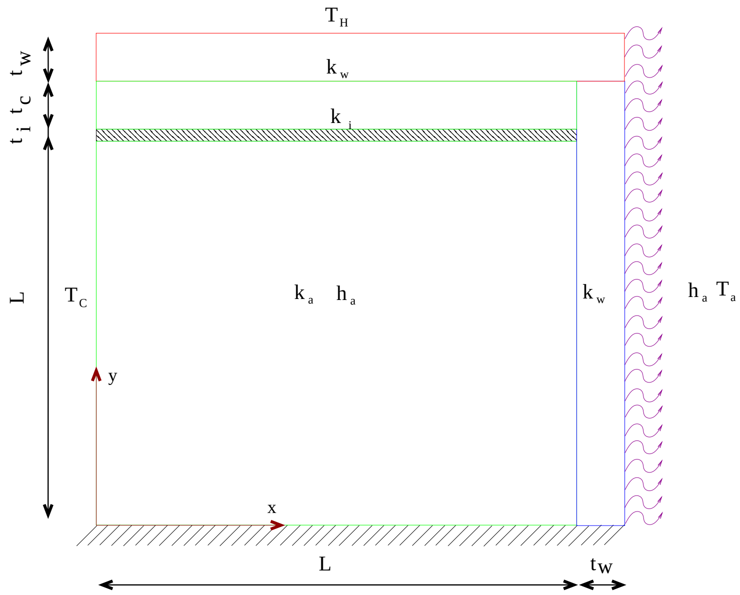


Figure 3.11: Boundary conditions of domain with insulation

More interesting observations can be drawn for the average heat loss across side and top wall for various conductivity ratios. As noted in the Tables 3.2, 3.2 and 3.4 representing that Q_{side} increases for the initial increase in the conductivity ratio for the thinner wall but it is interesting to note that the further increase of conductivity ratio causes the Q to decrease,this variation is clearer for the higher Reynolds number , contrary to that of the thick wall. The effect of Reynolds number over heat loss across side wall is more significant to that of the heat loss across top wall.

3.2 Heat transfer and fluid flow characteristics of square enclosure with insulation

Considering the similar parameters as for the case of without insulation a further study was carried out for studying the effect of insulation on the results by extending the same numerical code used to study And interesting observations from the results are discussed in the following section. Figure 3.11 represents the 2d model under consideration. As it is the extension of earlier study the boundary condition and governing equation are kept unchanged. The figure 3.11 shows that it has 5 blocks as compared to 3 blocks with the earlier model.

Table 3.4: Heat transfer and Nusselt number for wall thickness $t = 0.1$

$K_r \Downarrow$	Q_{side}			Q_{top}			$\overline{Nu}_{side}$			$\overline{Nu}_{top}$			θ_{avg}		
$Re \Rightarrow$	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4
10.417	2.979	7.001	9.705	1.307	1.68	2.21	4.67	12.29	18.52	1.048	-0.0047	-0.326	0.38	0.325	0.288
20.83	2.993	7.373	10.534	1.256	1.424	1.731	4.497	11.788	17.684	0.951	-0.626	-1.564	0.391	0.349	0.323
31.25	2.991	7.491	10.813	1.239	1.33	1.54	4.427	11.564	17.28	0.918	-0.863	-2.07	0.395	0.359	0.338
41.67	2.987	7.545	10.944	1.229	1.281	1.437	4.387	11.436	17.037	0.9	-0.989	-2.349	0.397	0.364	0.346
52.08	2.983	7.573	11.016	1.223	1.250	1.373	4.360	11.351	16.872	0.889	-1.067	-2.524	0.399	0.367	0.351
62.5	2.979	7.592	11.061	1.22	1.23	1.33	4.342	11.29	16.753	0.882	-1.12	-2.645	0.3999	0.37	0.355
72.917	2.976	7.603	11.09	1.216	1.216	1.297	4.327	11.246	16.662	0.876	-1.158	-2.734	0.401	0.372	0.358
83.33	2.972	7.611	11.11	1.213	1.208	1.273	4.316	11.211	16.591	0.871	-1.188	-2.801	0.401	0.373	0.36
100	2.969	7.62	11.132	1.21	1.192	1.244	4.302	11.169	16.503	0.865	-1.222	-2.882	0.402	0.374	0.362

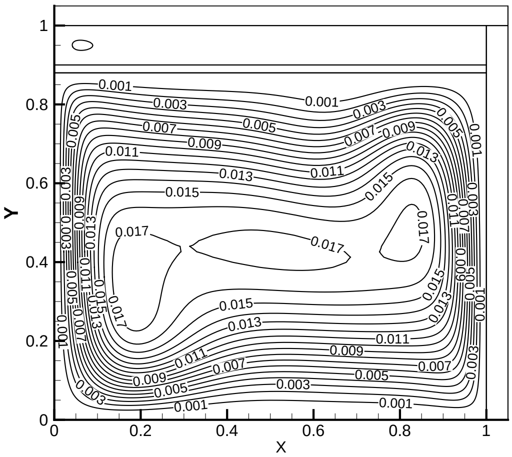
3.2.1 Flow Characteristics

Figure 3.13 shows a typical streamline contours for the case of an enclosure with an insulation wall. The flow of the air trapped between the insulation and the top wall is too weak. All the other observation are expectedly repeated as with earlier case.

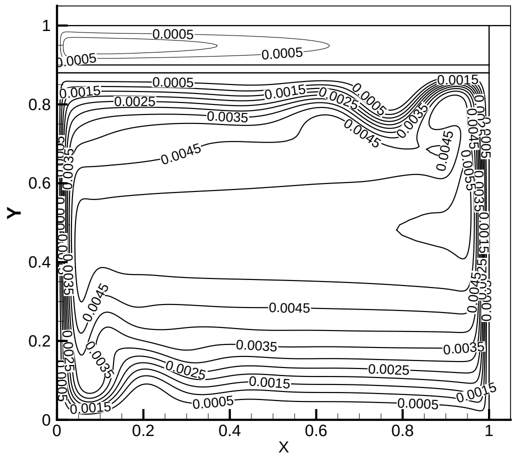
. It is interesting to observe that the presence of an insulation wall decreases the strength of convection inside the enclosure. As a result, the heat from the top does not get convected inside the enclosure which reflects in a lower average temperature of the enclosure. Also, a weak convection is induced in the trapped air inside the cavity above the insulating wall, the strength of which increases with the increase in the Reynolds number. Similar observation can also be noticed for the isotherms which is shown in figure 3.16.

3.2.2 Effect of Insulation near the top wall

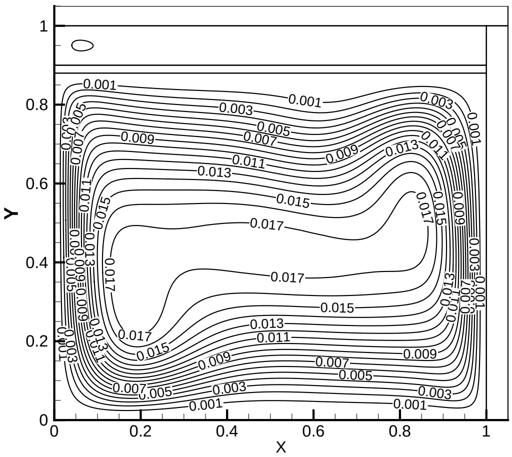
To study the effect of insulation, a low thermal conductivity wall is provided at a non-dimensional distance of 0.1 from the top. The results are compared and discussed in the following section. Due to the insulation most of the heat is constrained in the cavity above the insulation. This allows the enclosure to maintain at a lower temperature, i.e cooler environment can be maintained within the enclosure. Insulation causes disturbance to the flow near the hot wall and also due to low conductivity of the insulation the heat is trapped in the cavity above it. This can be noted in Tables 3.5, 3.6, and 3.7, where the comparison can be made for the average temperature for both the cases. To compare the average temperature based on equal volume of the enclosure, by neglecting the thickness of the insulation, the following method is adopted, overall $\theta^o_{avg} = \left(V_e\theta^e_{avg} + V_c\theta^c_{avg}\right)/(V_e+V_c)$, where θ^e_{avg} and θ^c_{avg} are average temperatures and V_e and V_c are volumes covered by the enclosure and the cavity respectively. It is quite obvious that although θ^o_{avg} is more compared to the average temperature of the enclosure without insulation, but the average temperature inside the enclosure, θ^e_{avg}, is lower by almost 10%. This suggests that due to the insulation, although lower temperature can be maintained within our interested area, the overall temperature of the whole domain is



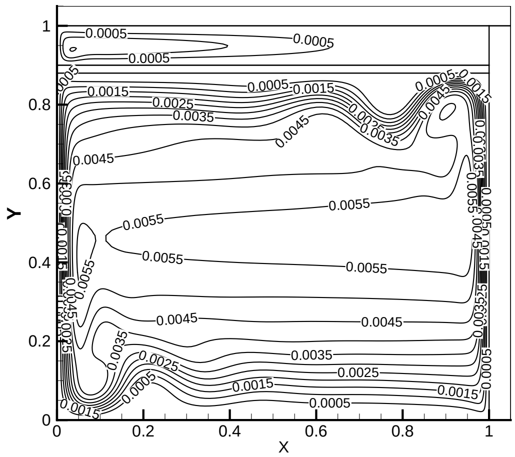
(a) K_r =10.417 and Re =10^3



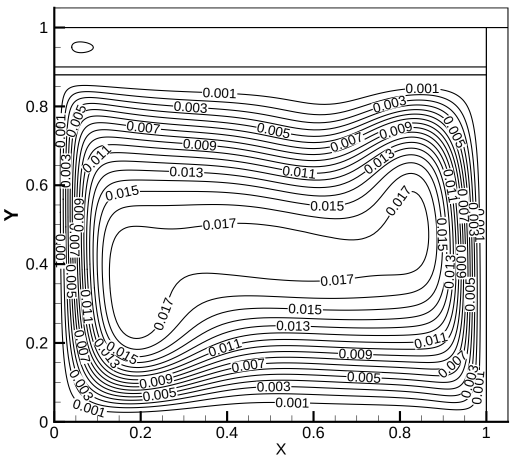
(b) K_r =10.417 and Re =10^4



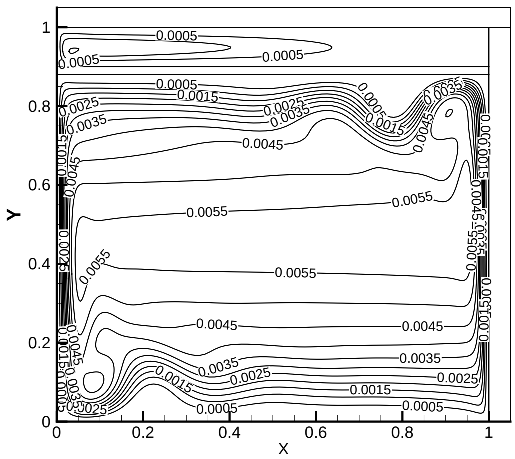
(c) K_r =41.67 and Re =10^3



(d) K_r =41.67 and Re =10^4

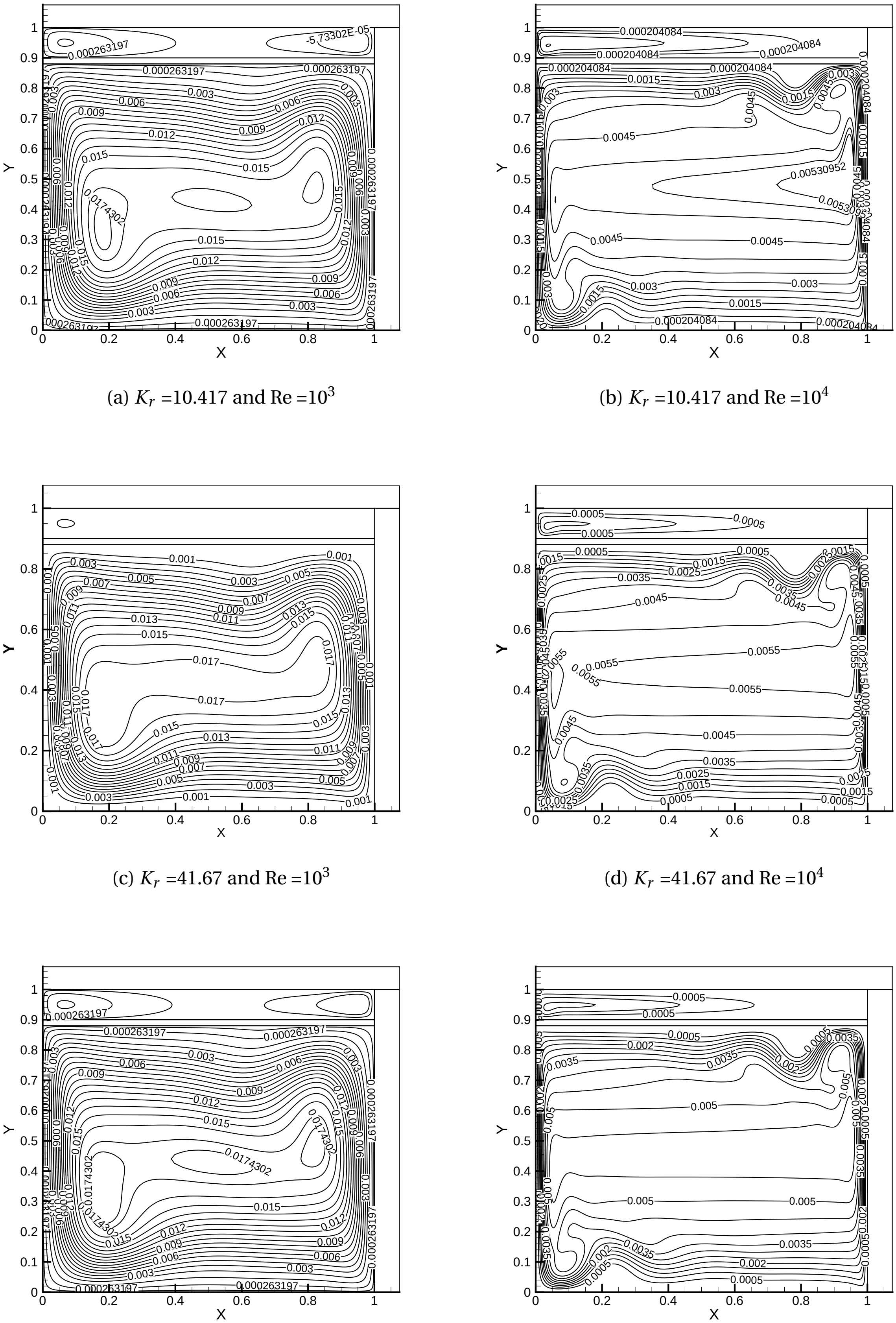


(e) K_r =100 and Re =10^3



(f) K_r =100 and Re =10^4

Figure 3.12: streamline plots for t=0.05 (with insulation wall)



(a) K_r =10.417 and Re =10^3

(b) K_r =10.417 and Re =10^4

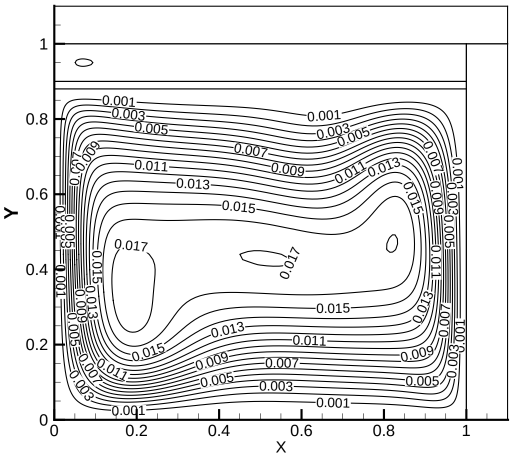
(c) K_r =41.67 and Re =10^3

(d) K_r =41.67 and Re =10^4

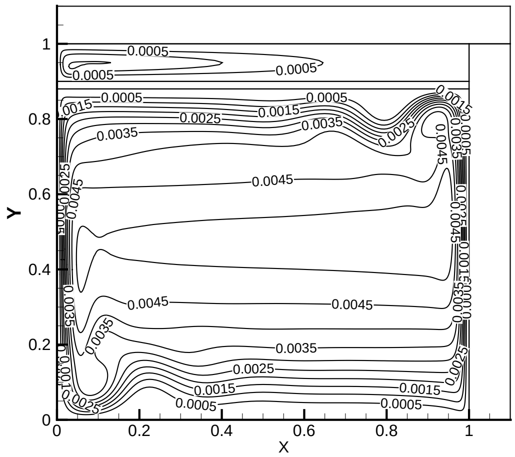
(e) K_r =100 and Re =10^3

(f) K_r =100 and Re =10^4

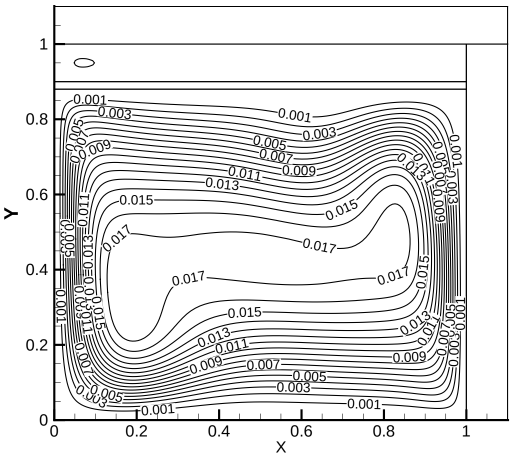
Figure 3.13: streamline plots for t=0.075 (with insulation wall)



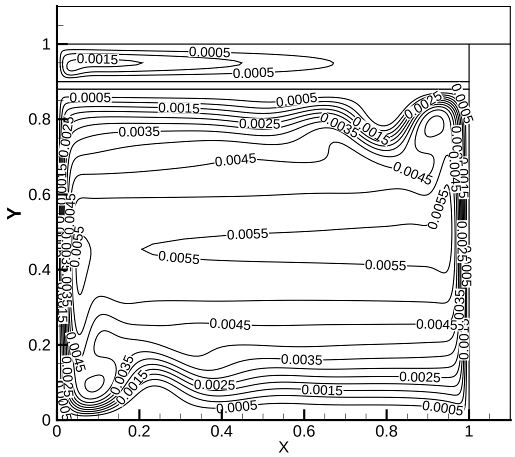
(a) K_r =10.417 and Re =10^3



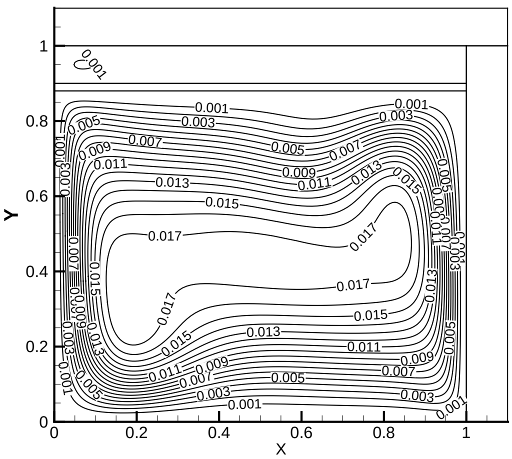
(b) K_r =10.417 and Re =10^4



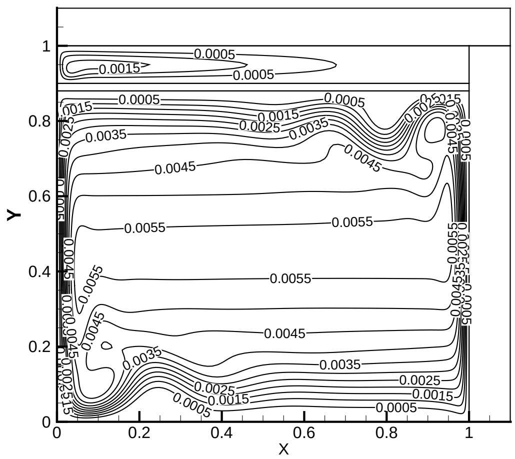
(c) K_r =41.67 and Re =10^3



(d) K_r =41.67 and Re =10^4

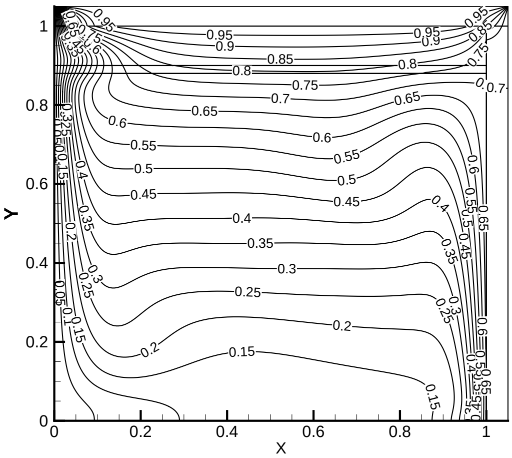


(e) K_r =100 and Re =10^3

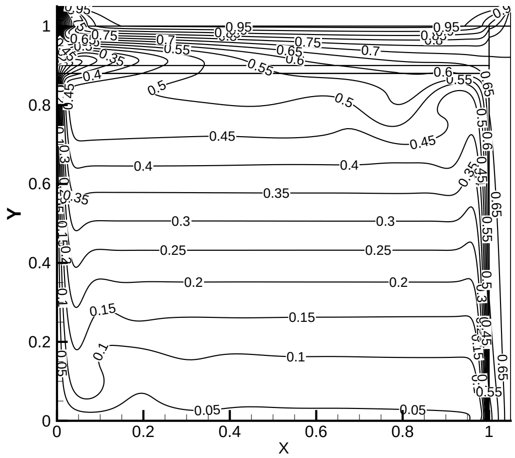


(f) K_r =100 and Re =10^4

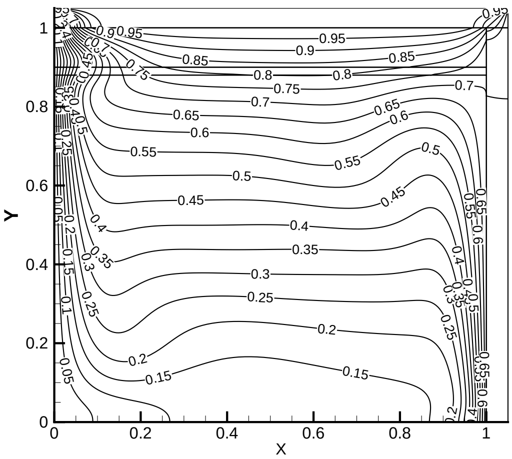
Figure 3.14: streamline plots for t=0.1 (with insulation wall)



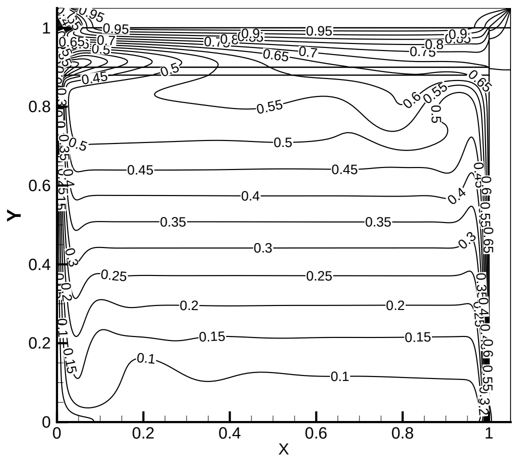
(a) K_r =10.417 and Re =10^3



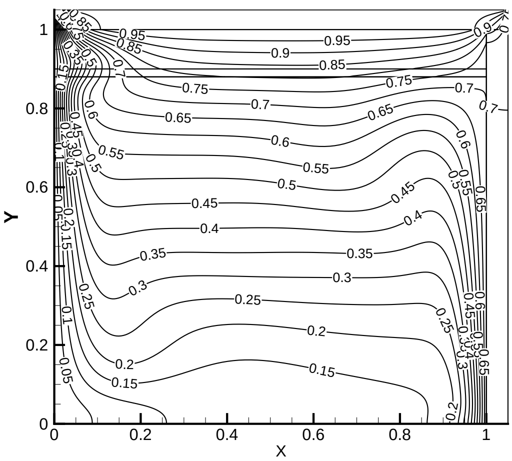
(b) K_r =10.417 and Re =10^4



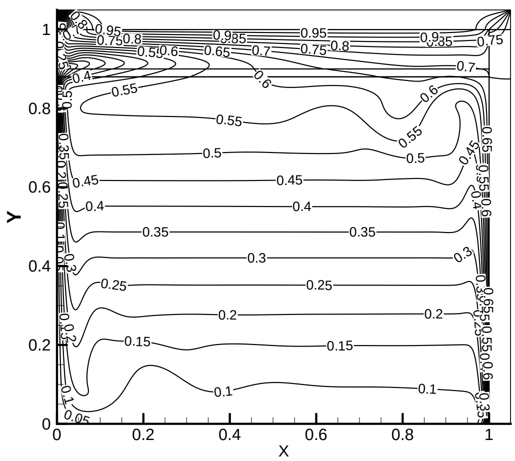
(c) K_r =41.67 and Re =10^3



(d) K_r =41.67 and Re =10^4

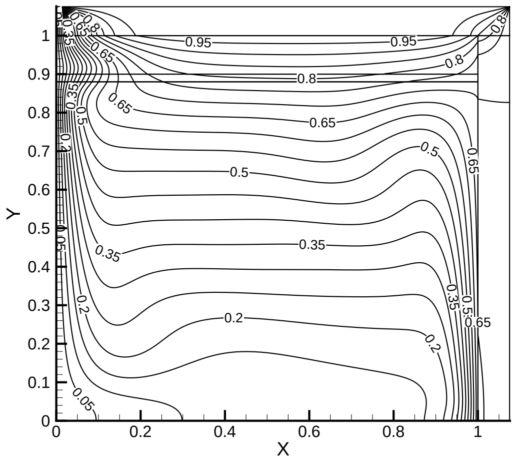


(e) K_r =100 and Re =10^3

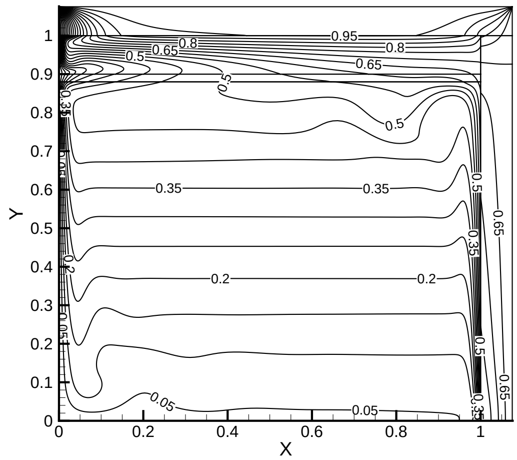


(f) K_r =100 and Re =10^4

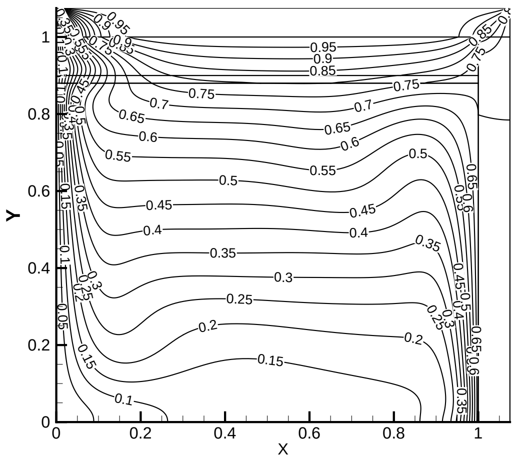
Figure 3.15: Isotherm plots for t=0.05 (with insulation wall)



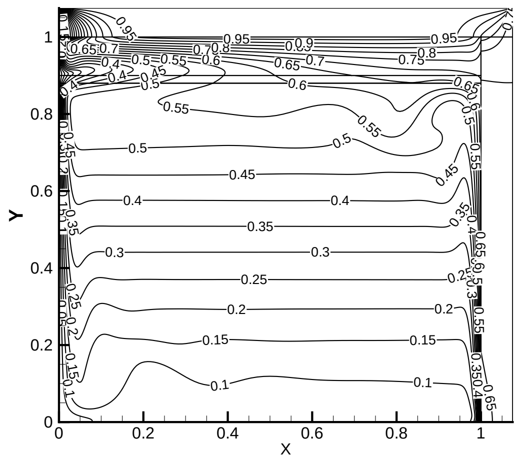
(a) K_r =10.417 and Re =10^3



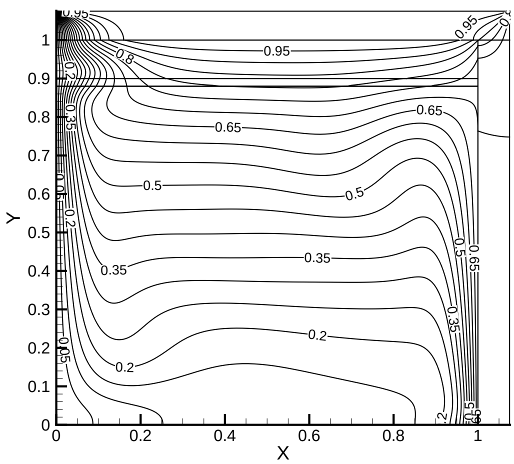
(b) K_r =10.417 and Re =10^4



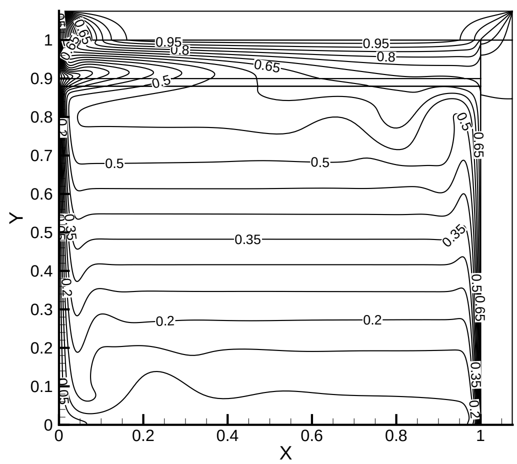
(c) K_r =41.67 and Re =10^3



(d) K_r =41.67 and Re =10^4

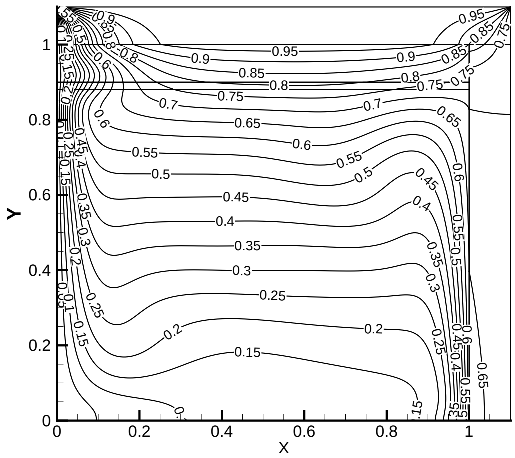


(e) K_r =100 and Re =10^3

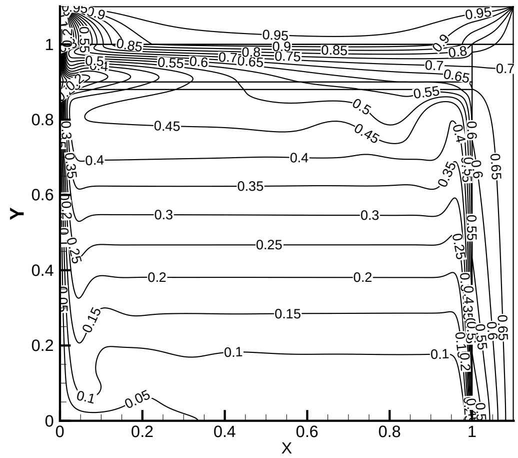


(f) K_r =100 and Re =10^4

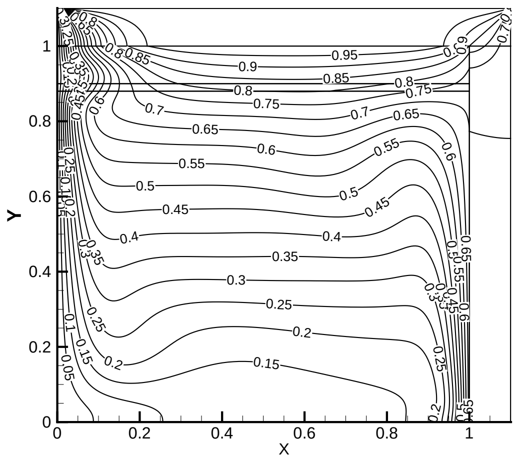
Figure 3.16: Isotherm plots for t=0.075 (with insulation wall)



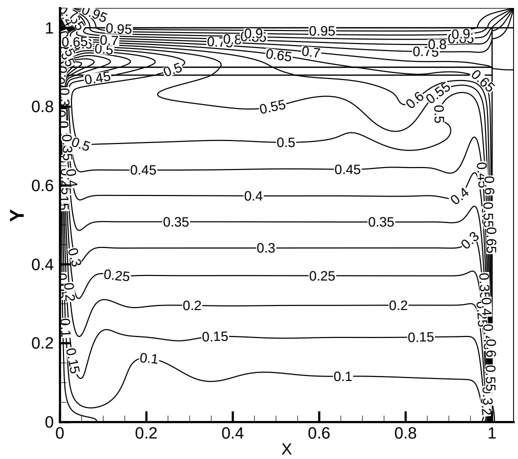
(a) K_r =10.417 and Re =10^3



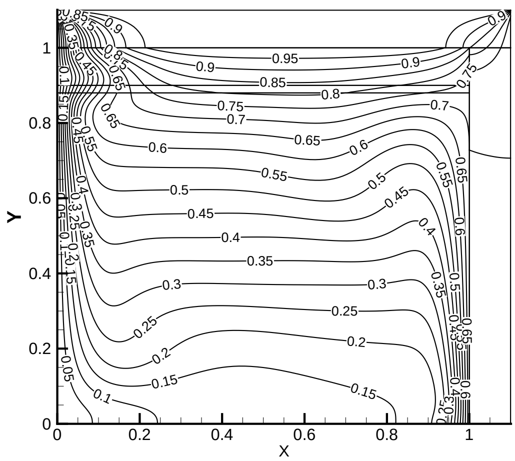
(b) K_r =10.417 and Re =10^4



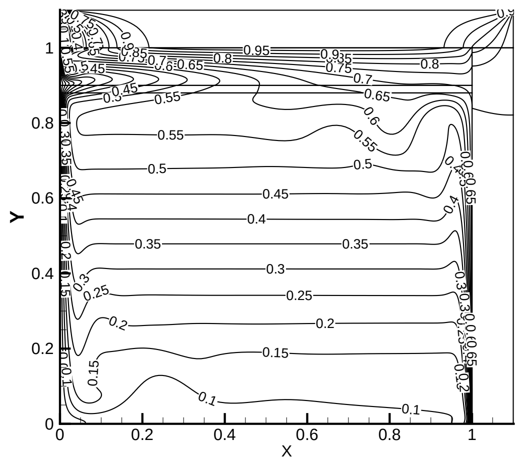
(c) K_r =41.67 and Re =10^3



(d) K_r =41.67 and Re =10^4



(e) K_r =100 and Re =10^3



(f) K_r =100 and Re =10^4

Figure 3.17: Isotherm plots for t=0.1 (with insulation wall)

more. This difference in the average temperature is notably more for the lower Reynolds number as can be seen in Table 3.7.

CASE⇒	With Insulation									Without Insulation		
$K_r \Downarrow$	θ^e_{avg}			θ^c_{avg}			θ^o_{avg}			θ_{avg}		
$Re \Rightarrow$	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4
10.417	0.366	0.309	0.272	0.832	0.750	0.687	0.414	0.354	0.314	0.392	0.346	0.312
20.83	0.371	0.324	0.295	0.838	0.761	0.704	0.419	0.369	0.337	0.397	0.361	0.335
31.25	0.373	0.331	0.305	0.839	0.766	0.711	0.420	0.375	0.347	0.399	0.366	0.345
41.67	0.374	0.334	0.311	0.840	0.768	0.714	0.421	0.378	0.352	0.401	0.37	0.350
52.08	0.375	0.336	0.315	0.841	0.769	0.717	0.422	0.381	0.356	0.402	0.372	0.354
62.5	0.375	0.338	0.318	0.841	0.771	0.718	0.422	0.382	0.359	0.402	0.373	0.356
72.917	0.375	0.339	0.321	0.842	0.772	0.72	0.423	0.384	0.362	0.403	0.374	0.358
83.33	0.376	0.341	0.323	0.842	0.772	0.721	0.423	0.385	0.363	0.403	0.375	0.359
100	0.376	0.342	0.325	0.842	0.773	0.722	0.423	0.386	0.366	0.404	0.376	0.362

Table 3.5: Average temperature of the enclosure for both the cases and wall thickness $t = 0.05$

CASE⇒	With Insulation									Without Insulation		
$K_r \Downarrow$	θ^e_{avg}			θ^c_{avg}			θ^o_{avg}			θ_{avg}		
$Re \Rightarrow$	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4
10.417	0.361	0.3	0.26	0.818	0.732	0.664	0.407	0.344	0.301	0.386	0.334	0.299
20.83	0.368	0.321	0.290	0.825	0.748	0.689	0.415	0.364	0.331	0.394	0.355	0.329
31.25	0.371	0.329	0.304	0.827	0.754	0.698	0.418	0.373	0.344	0.397	0.363	0.341
41.67	0.373	0.334	0.312	0.828	0.757	0.703	0.419	0.377	0.352	0.399	0.367	0.348
52.08	0.374	0.337	0.317	0.829	0.759	0.706	0.420	0.381	0.357	0.4	0.37	0.353
62.5	0.374	0.339	0.321	0.830	0.761	0.708	0.421	0.382	0.360	0.401	0.372	0.356
72.917	0.375	0.341	0.324	0.831	0.762	0.710	0.422	0.384	0.363	0.402	0.373	0.358
83.33	0.376	0.343	0.326	0.831	0.763	0.711	0.422	0.386	0.365	0.402	0.374	0.36
100	0.377	0.345	0.328	0.832	0.765	0.713	0.423	0.387	0.368	0.403	0.375	0.362

Table 3.6: Average temperature of the enclosure for both the cases and wall thickness $t = 0.075$

CASE⇒	With Insulation									Without Insulation		
K_r⇓	θ^e_{avg}			θ^c_{avg}			θ^o_{avg}			θ_{avg}		
Re ⇒	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4
10.417	0.357	0.292	0.251	0.802	0.715	0.644	0.402	0.335	0.292	0.38	0.325	0.288
20.83	0.367	0.318	0.289	0.812	0.735	0.675	0.412	0.361	0.328	0.391	0.349	0.323
31.25	0.371	0.329	0.304	0.815	0.743	0.686	0.416	0.371	0.343	0.395	0.359	0.338
41.67	0.373	0.335	0.313	0.817	0.747	0.692	0.418	0.377	0.352	0.397	0.364	0.346
52.08	0.374	0.339	0.319	0.818	0.749	0.695	0.419	0.380	0.357	0.399	0.367	0.351
62.5	0.375	0.341	0.323	0.819	0.751	0.698	0.421	0.383	0.361	0.3999	0.37	0.355
72.917	0.376	0.343	0.326	0.82	0.753	0.7	0.421	0.385	0.364	0.401	0.372	0.358
83.33	0.377	0.345	0.329	0.821	0.754	0.701	0.422	0.386	0.367	0.401	0.373	0.36
100	0.378	0.346	0.332	0.821	0.755	0.703	0.423	0.388	0.369	0.402	0.374	0.362

Table 3.7: Average temperature of the enclosure for both the cases and wall thickness $t = 0.1$

CHAPTER 4

Conclusion, limitation and future work

4.1 CONCLUSION

Numerical simulation are carried out for steady two dimensional laminar natural convection flow of air in a square enclosure and the results are discussed in the preceding section. various parameters are studied includes Reynolds number, wall thickness, and conductance ratio to investigate the behaviors on the temperature and flow field. From the above outlined result following conclusion is drawn:

1. The thickness of the wall of the enclosure should not be more as the result shows the average temperature is maintained constant for various thickness for high Reynolds number and high conductance ratio, so the average temperature can be maintained in the enclosure by using optimum thickness of the wall.

2. The effect of boundary layer is observed in the high Reynolds number as compared with the low Reynolds number as the driving force increases with the Reynolds number.

3. But as the Reynolds number increases the disturbances could be observed in the enclosure. The perturbations are observed in the streamlines plots at bottom left and

top right corner of the enclosure suggesting more effective zone to carry any operation there.

4. Due to installation of insulation, lower temperature could be maintained in the enclosure. Also the heat at the top cavity could be used for other applications like storage or drying purposes. Insulation provided at the given height helps to reduce the temperature by almost 10% which is significant value of decrease in today's competitive world as it will decrease the load on cooling devices.

4.2 Limitations and Recommendations

Although the current numerical experiment is exhaustive, this study is conducted by considering only a simple case of steady laminar flow. A similar exhaustive study for the case of turbulent flow will be of more important practically. As in practical a normal size room will always have a natural turbulent flow. This study is just a scratch on the surface for this field. One can study further by gradually increasing the complexity of the problem by involving the various factors occurring in actual practice such as the various layers on the wall or ceiling of concrete, bricks, tile, facade etc**.**

A transient analysis of one entire day should be done considering a unsteady condition. It will give us a clear idea of the temperature variation inside the enclosure. The effect of the forced flow on the temperature and flow field should be studied. Also the the effect of vents at various position will give us interesting result to study and compare with the current study. As the radiation is also an important mode of heat transfer at high temperature its effect might prove significant when higher temperature is to be considered at the wall. So the including radiation model is another factor to be considered during future work. The domain may be complex according to the application so studying the complex geometry will be beneficial.

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